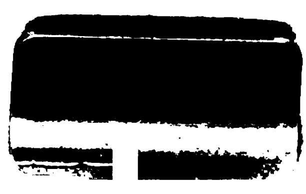

INTERNAL COMBUSTION ENGINES

INTERNAL COMBUSTION ENGINES

THEORY AND DESIGN

A TEXT BOOK ON GAS- AND OIL-ENGINES FOR ENGINEERS AND STUDENTS IN ENGINEERING

BY

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PREFACE

A LARGE part of the material in this book, especially in the portion dealing with design, has been used for several years by the author in the form of notes for a course in gas- and oil-engine design. These notes were added to from time to time until they became too bulky for use in typewritten form. The first thought of the author was to have the notes printed as such for private use only but it was finally decided to publish them in the form of a text book. In order to do this it was necessary to expand them and in doing this certain investigations and writings of other authors were consulted and drawn on for material without which a book of this nature would be valueless.

The book is written with the assumption on the part of the author that the reader has studied thermodynamics and mechanics. However, the author thought best to include in the book a review of the parts of thermodynamics which are absolutely essential to a clear understanding of the theory of internal combustion engines. Also, at certain places in the book, the author has digressed far enough to derive and explain certain indispensable relations in mechanics, not that they present anything new but because of a possibility that the reader has allowed them to become hazy from lack of use.

The author wishes to express his thanks to those manufacturers who kindly furnished drawings, photographs and data and to those publishers and authors who permitted their work to be used in the text of the book. He also expresses his thanks to his associates who assisted him in the preparation of drawings and particularly to Professor A. M. Greene, Jr., for his valuable aid and advice on portions of the book.

R. L. S.

RENSSELAER POLYTECHNIC INSTITUTE,
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CONTENTS

CHAPTER XII

CHAPTER XXIII

INTERNAL COMBUSTION ENGINES

CHAPTER I

GENERAL

In writing this book the author assumes that it is to be used primarily as a text book, and that the student using it has a knowledge of mechanics and thermodynamics. Many students, however, who have a working knowledge of the subjects mentioned above are not at all familiar with the mechanical details of gas- and steam-engines. This chapter is devoted to giving the student a general idea of the details of gas-engines, the names and functions of the various parts and the differences that exist between the gas-engine and the steam-engine.

One other assumption made by the author might well be explained at this point. No reference is made to the preparation of any kind of solid fuels in order to use them in gas-engines. In speaking of fuels for gas-engines, then, it is with the assumption that the fuel is in the gaseous form, ready to be introduced into the engine.

Conversion of Energy.—The conversion of **chemical energy** into **mechanical energy** is accomplished by means of an engine, steam or gas. If a steam-engine is used, an intermediate member is placed in the chain, this member being the boiler. The **chemical energy** of the fuel is changed into **heat energy** in the furnace and this heat energy is transferred to the water in the boiler. The **water** in this case acts as a carrier and does not enter vitally into the whole problem of heat conversion by means of the steam-engine. That is to say, other carriers could be used in place of water, but it happens that water is plentiful and has certain characteristics which make its use as such a carrier desirable in many ways.

In the **engine cylinder** the heat in the steam is converted into **mechanical energy,** by moving a piston to and fro in this cyl-

inder. In some cases the reciprocating motion is used as such; in the great majority of engines, however, the reciprocating motion is changed into a rotary motion by means of a system of members forming a kinematic chain.

In the internal combustion engine **combustion** takes place directly in the **cylinder,** that is, the chemical energy is changed into heat energy in the cylinder, which acts, in this case, as the furnace of the boiler.

The **medium** or **carrier** in the internal combustion engine is formed from the **fuel** and the **air** required for combustion. This acts very nearly as a **perfect gas,** while the action of the steam in a steam-engine cylinder is far from that of a perfect gas. The mechanical action of the steam-engine is very similar to the internal combustion engine.

Advantages of a Gas-engine. — If the actions of a steam-engine and an internal combustion engine are almost identical, why use one in preference to the other? The answer, theoretically, is this: — The **efficiency** of the Carnot cycle, the highest attainable, is $\frac{T_1 - T_2}{T_1}$, where T_1 is the absolute temperature of the high point of the cycle and T_2 is the temperature of the low point, or, in other words, the cycle works between limits of T_1 and T_2. The value of T_2 is fixed by the atmospheric temperature and is the same for both the internal combustion engine and the steam-engine, assuming both these engines to operate on the Carnot cylinder. In the steam-engine, the value of T_1 is about 862 degrees for 250 pounds steam pressure. With a value of T_2 at 70° F. or 530 degrees absolute the efficiency would be 0.385. With a temperature of 2500° F. in the cylinder of the gas-engine the efficiency would be 0.83 or more than twice that of a steam-engine. Possibly, then, the gas-engine has the advantage of the steam-engine in the ratio of 0.83 to 0.385; but theoretical considerations alter this considerably. These considerations will be treated later in this book.

Heat Losses. — In the **steam-engine cycle** if the steam is cooled below the boiling point corresponding to its pressure some of it is condensed and the heat radiated is a direct loss. The problem, then, is to retain all the heat of the steam until the steam leaves the cylinder, or at least to prevent any radiation, so that the maximum amount of heat will be available for work.

Another, and greater loss, is the **heat** of **vaporization** at exhaust pressure. A large part of the heat that is transferred from the boiler furnace into the water in the boiler goes into the latent heat, or heat of vaporization. This heat is not available for doing work but is lost by being discharged into the atmosphere.

With the gas-engine the reverse is true and, instead of retaining all the heat in the cylinder, some of it must be removed by conduction because temperatures are high enough to be injurious to the metal in contact with the hot gases. This excess heat is removed by means of jacket water.

The effect of **heat** on the gases in a gas-engine cylinder is to increase the **pressure.** In order to reduce T_2 as low as possible and thus increase the efficiency the gases are expanded in the cylinder. To bring the gases in the cylinder down to atmospheric temperature **complete expansion,** or expansion down to zero pressure, would be necessary. Practically, that is not possible and the gases are exhausted at a pressure and temperature considerably above the temperature of the atmosphere. This is another large loss of heat which helps to decrease the theoretical efficiency.

Source of Power. — Inflammable **gas** or vapor is the source of heat in the gas-engine cycle. The **gas** is **ignited** in the cylinder, **temperature increases** and the resulting **pressure** drives the **piston** forward. Many different kinds of gas, varying in heating value, are employed, and the effects obtained by ignition and explosion cannot be determined or understood without a knowledge of the chemical constituents of the gas, and the proportions in which they combine with the oxygen of the air. Since the gas does not contain the oxygen necessary for combustion, it can never be burned by itself, but must always be diluted with a certain quantity of air. Unless the composition of the gas and the ratio of its dilution with air are known, it is impossible to ascertain the temperatures and pressures in the cylinder and to calculate the work the gas ought to do. The value in design of the laws and phenomena of the combustion of gas and air mixtures will be taken up later in the part of this book devoted to the design.

Utilization of Power. — A brief study of the **construction** and **parts** of a gas-engine is necessary in order to become familiar with the engine as a whole and the terms used in connection with

the various parts. We will enumerate these parts and then describe the functions they have to perform, together with the different operations taking place in a gas engine.

Base. — The **base-plate,** on which the engine is fixed and the cylinder bolted, is of **cast-iron,** usually, and of massive construction. This base-plate appears in a variety of forms, varying with the type of engine, use to which the engine is to be put and the manufacturer.

Cylinder. — The gas-engine **cylinder,** made usually of **cast-iron,** is either horizontal or vertical, depending on the type of engine. In small engines only one single-acting cylinder is used, one end being open to the atmosphere. In larger engines two or more single- or double-acting cylinders are used. As one object to be attained in a gas-engine is to have the gases expand to a low pressure, it appears that if the engine is made compound this object will be attained. This arrangement, though often tried, has never been found to be successful.

Gas can be **ignited** in an engine cylinder **at atmospheric pressure,** but as higher pressures and greater efficiency result if the gas is **compressed** previous to ignition, this method is used. In almost every case the gas is compressed in the working cylinder, one stroke of the cycle being devoted to this work.

A special feature of gas-engine cylinders is that, on account of **high temperatures** reached, they are always provided with some means of **cooling** the walls. In the smallest sizes it has been found sufficient to increase the radiating surface by making the outer surfaces of the walls ribbed, exposing a large cooling area to the outside air. In larger engines this "air-cooling" is not sufficient, so the cylinders are cast with double walls, between which water is circulated to keep the inner wall cool. In single-acting engines the cooling effect is increased by the open end, the motion of the piston causing currents of air to rush into the cylinder at every revolution, thus greatly increasing the amount of heat conducted away from the piston.

Pistons. — The **pistons** of gas-engines are very similar to steam-engine pistons except that they are made longer. In single-acting engines **trunk pistons** are used. Usually no special provision is made for cooling this type of piston as the circulation of air within the piston is very efficient. In **double-acting** engines, however, the pistons are cast hollow and are water-

cooled. All pistons are provided with Ramsbottom or snap rings to prevent leakage between the piston and cylinder walls.

Valves. — The **valves** of a gas-engine perform the same operations that the valves of a steam-engine do. They admit fresh gas into the cylinder at the proper time and allow the exhaust gases to escape. In addition, in some types of engines they assist in mixing the gases and in timing the explosions. In some of the earlier engines **slide-valves** were used to perform these functions, but as temperatures and pressures increased, the limitations of this type of valve were soon reached and poppet or lift valves were substituted. Occasionally, the fresh gas is admitted to the cylinder through cylindrical or piston valves but their use is not general.

In most gas-engines valves are worked from a **side-shaft** by **cams** or **eccentrics**; in others they are automatically lifted or closed by the pressure in the cylinder.

Transmission of Energy. — As in a steam-engine the force on the reciprocating gas-engine piston is transmitted to the revolving **crank-shaft** through a **connecting-rod** and **crank.** In single-acting engines there is no piston-rod but the connecting-rod is pinned directly to the trunk piston which also acts as a crosshead. In double-acting engines the arrangement is identical with that of a steam-engine, there being a **piston-rod, crosshead** and **connecting-rod.**

In all gas-engines five operations are required for a complete cycle:

I. Admission and mixture of the gas and air.
II. Ignition.
III. Explosion or combustion.
IV. Expansion.
V. Exhaust, or discharge of the burned or inert gases.

To these another has been added in modern engines, namely, compression.

This **cycle** of work corresponds to one explosion but not necessarily to one revolution. In many engines the number of explosions and revolutions are independent of each other. The different cycles will be analyzed in a following chapter.

Admission of Gas and Air to the Cylinder. — In modern engines **admission** is affected with ordinary **lift valves,** both gas and air passing through the same valve. Before entering the

cylinder the gas usually passes through a chamber where it is thoroughly mixed with the proper proportion of air, which is admitted to the mixing chamber through a separate inlet. The valve which admits gas into the mixing chamber is usually connected to the **governor,** which regulates the quantity of gas entering, and consequently the number or strength of the explosions.

Ignition. — In the older types of engines **ignition** of the charge was done with an **open flame** which was located in a cavity separated from the cylinder by a valve, usually a slide-valve. At the proper time for ignition this slide-valve opened and put the flame into communication with the compressed charge in the cylinder proper. The explosion that followed usually extinguished the flame. To remedy this trouble a second valve was used to put this flame chamber into communication with a permanent flame outside the cylinder after the slide-valve leading from the flame chamber into the cylinder had closed.

The **flame ignition** has been superseded by others which are more reliable, chief among these being the **hot-tube, electric spark** and **combustion due to high compression temperatures.** The **hot-tube** ignition is similar to the flame in that it is placed in a separate chamber communicating with the cylinder proper by a valve which opens at the proper time. The tube which is closed at its inner end is kept hot by an external flame. The **electric ignition** is by means of a current from **primary batteries** or from a **magneto** or **dynamo.** The two systems of electric ignition, the "**make and break**" and the "**jump spark,**" will be dealt with later. Ignition by **high compression temperature** is used in high-pressure oil-engines where air is compressed to such a high pressure that the resulting temperature is above the ignition point of the fuel oil. Then when the fuel is sprayed into the cylinder, spontaneous combustion results. Obviously this system of ignition cannot be used with gaseous fuels.

Explosion and Expansion. — **Explosion** of the charge takes place in the combustion space of the cylinder as ignition occurs. In one type of engine that is now obsolete, ignition, and consequently explosion, took place when the piston had completed about one-half its suction stroke. Ignition thus occurred at atmospheric pressure. In order to increase the efficiency, compression was introduced. The pressure to which compression is now carried determines the relative volume of the combustion

or clearance space. The **point of ignition** in modern engines is determined by practical consideration but is usually at or near the dead center, or, in other words, when the piston is at or near the inner end of its stroke.

Exhaust. — Various methods have been employed to get rid of the **burned gases** after explosion, but practice has shown that they should be expelled from the cylinder as quickly and as completely as possible. In single-acting engines the exhaust-valve is closed during the forward suction stroke of the piston. In some engines it is open during half the return exhaust stroke, in others during the whole of this stroke, and in still others it is open during the last part of the forward stroke and the first part of the return stroke. In this last case the new charge is forced into the cylinder under pressure, displacing the burned gases.

The **exhaust-valve** of the modern gas-engine works under severe conditions, as it opens when the pressure and the temperature are high. On account of this the heat lost in the exhaust is considerable. This is one of the defects of the modern engine which engineers are anxious to remedy.

Compression. — The one **improvement** which has done more than any other to make the modern gas engine successful is higher **compression.** It is done as follows: A charge of air and gas is drawn or forced into the cylinder and on the **return stroke** this charge is compressed into the **combustion space.** The volume of the combustion space is determined by the intensity of the compression. The advantage of this process is that the particles of gas are pressed more firmly together so that when ignition takes place the resulting temperature and pressure is higher than if the gas had not been compressed.

The advantages of compression are:

I. The **smaller size of cylinder** required. Since the pressure resulting from combustion is higher with compression than without, it follows that the total force on the piston is greater, consequently the work done is greater.

II. **Combustion** is more certain and more rapid than without compression because the particles of gas are forced close together, and ignition proceeds rapidly as the heat is communicated more rapidly from one particle to another.

III. **Greater economy** of gas because the resulting temperature is higher. It was shown earlier in this chapter that the

efficiency increased as the maximum temperature reached increases.

IV. **Poorer gas** can be used than without compression because combustion takes place so rapidly that the poor grade of gas will explode, whereas with no compression combustion would be so slow as to prevent the use of that gas in an engine.

Oiling. — The question of **lubrication** is of very great importance in a gas-engine. Not only do we have the usual crank-pin, wrist-pin and shaft bearings to lubricate, but the piston requires the utmost care in this respect. In the steam-engine cylinder lubrication is not particularly difficult on account of the surfaces being wetted and the low temperature. In the gas-engine, however, the rubbing surfaces are dry and the temperature and pressure are both high. On account of the high temperature the lubricating oil for the gas-engine must be carefully selected.

Starting. — A **steam-engine** is started by turning the steam under pressure into the cylinder. A gas-engine cannot be started in this way because there is no pressure to be turned into the cylinder unless compressed air be used. **Large engines** are started with **compressed air** in exactly the same way that steam is used in a steam-engine. Small engines are set in motion by hand or by an electric motor, this operation continuing until explosions start in the cylinder.

Speed Regulation. — **Speed regulation** is accomplished in one or two of several ways in a gas-engine:

I. The **amount** of **mixture** may be cut down or increased to suit the load. This is **throttling.**

II. The **ratio** of **gas to air** may be changed, thus changing the power of the engine to suit the load. This is **constant quantity** governing.

III. The **mixture** may be **cut off** entirely for several working strokes in succession, the number depending on the load, thus preventing explosions or impulses for these strokes. This is the "**hit-and-miss**" method and is used only on small engines. The cutting off of the supply of mixture may be accomplished in various ways which will not be discussed at present. No matter what system of governing is used, the change of centrifugal force of a revolving weight, or the change of inertia of a weight, due to the change in speed, is used to actuate the mechanism which makes the proper change in speed, back to normal.

CHAPTER II

HISTORICAL

General. — Among the earliest attempts to obtain motive power from heat in an internal combustion engine was to ignite inflammable powder and utilize the force of the explosion to drive out a piston or the vacuum resulting from the cooling of the gases after the explosion to draw in a piston. The cannon was, strictly speaking, the first heat motor, the ball being the piston. The chief objection to using the cannon for developing power to drive machinery was that a new piston was necessary for each stroke, for there was no such thing as a return stroke.

Hautefeuille. — The first man to propose the use of an explosive powder to obtain power was the Abbé Hautefeuille, a son of a baker, at Orleans. He designed not only the first engine worthy of a name, but the first machine to use heat as a motive force, and capable of producing a definite quantity of continuous work. In 1678 he suggested the construction of a powder motor to raise water. The powder was burned in a vessel communicating with a reservoir of water. As the gases were cooled after combustion a partial vacuum was formed, and the water was raised by the atmospheric pressure from the reservoir. Another machine described by him was based on the principle of the circulation of the blood, produced by the alternate contraction and expansion of the heart. This was the first instance of a direct-acting machine but no machine could be made strong enough to withstand the spasmodic expansion of the powder as here proposed.

Huyghens. — Hautefeuille does not seem to have actually constructed any of the engines which he designed; but **Huyghens,** who was the first, in 1680, to employ a cylinder and piston, constructed a working engine and exhibited it to the French Minister of Finance. The powder in this engine was ignited in a little receptacle screwed on the bottom of the cylinder. The latter was immediately filled with flame, and the air in it was

driven out through leather tubes, which by their expansion acted, for the moment, as valves. The piston was forced by the pressure of the atmosphere into the vacuum thus formed. Huyghens found some difficulty in making the leather valves work, and in 1690 an attempt was made by **Papin** to improve upon the principle. By providing the valves with hydraulic joints Papin contrived to make them tighter and to obtain a better vacuum; but he found that, in spite of all that he could do, a fifth part of the air still remained in the cylinder and checked the free descent of the piston. After various attempts had been made to overcome this difficulty, he abandoned the use of explosive powder and devoted his attention to steam.

Barber. — For more than 100 years after these early attempts nothing was done in the way of making an internal combustion engine but the energies of the engineers and inventors were all turned towards the steam-engine and its application to the production of power. Gas extracted from coal had not yet been applied as a motive power in engines, and experience had shown that explosive powders were too dangerous to do much experimenting with. The first to design and construct an actual gas-engine was **John Barber,** who took out a patent in 1791. Barber made the gas required by his engine from wood, coal, oil or other substances, heated in a retort, from whence the gases obtained were conveyed into a receiver and cooled. A pump next forced them, mixed with the proper amount of atmospheric air, into a vessel called the "exploder." Here they were ignited and the mixture issued out in a continuous stream of flame against the vanes of a paddle wheel, driving them around with great force. Water was also injected into the explosive mixture to cool off the mouth of the vessel.

Barber's engine exhibits in an elementary form the principle of what is now known as combustion at constant pressure. It also resembles, crudely, one form of gas-turbine that is being investigated today.

Street. — The next engine, invented by **Robert Street,** in 1794, was a step in advance. Inflammable gas was exploded in a cylinder and drove up a piston by its expansion, affording the first example of a practical internal combustion engine. The gas was obtained by sprinkling spirits of turpentine or petroleum in the bottom of the cylinder and evaporating it by a

fire beneath. The upstroke of the piston admitted a certain quantity of air, which mixed with the inflammable vapor. Flame was next sucked in from a light outside the cylinder, through a valve uncovered by the piston, and the mixture of gas and air ignited. The explosion drove the piston up and forced down the piston of a pump for raising water. In this engine many modern ideas were foreshadowed, such as ignition by the external flame and the drawing in of the gas on the suction stroke of the piston; but the mechanical details were crude, indeed.

Lebon. — A great improvement in an engine, the patent for which was obtained in 1801, was made by **Lebon.** The machine was double-acting and the explosion of gas took place alternately on each side of the piston. The piston-rod operated the motor shaft as well as two pumps, in which gas and air were compressed, before they entered the motor cylinder. The inventor proposed to fire the mixture by an electric spark and suggested that the machine generating the spark should be driven from the motor shaft. The excellent theoretical principles on which the machine had been designed were striking for that early period and marked a new era in gas-engines. The progress of the internal combustion engines was probably retarded many years by the assassination of this skillful engineer in 1804. It was many years before the advantages of many principles which Lebon clearly understood were fully realized.

Lenoir. — From the time of Lebon, 1801, until 1860, many inventors worked on the application of gas to generating power, but no one was successful in getting a practical machine. The usual type of machine designed or built in that period was composed of a free piston in a vertical cylinder, open at the top end. A rack on the piston engaged a gear on the shaft above the cylinder. When the gas exploded, the piston was forced upward, rising freely on account of the ratchet connection of the gear on the shaft. As the gases in the cylinder cooled, the pressure dropped below atmosphere and the piston returned to the bottom of the cylinder, turning the shaft, through the ratchet, by its weight.

To **Lenoir** belongs the honor of building the first gas-engines that were used commercially for the production of power, engines which were built and sold as a business proposition; engines that for a time were immensely popular and gave good satis-

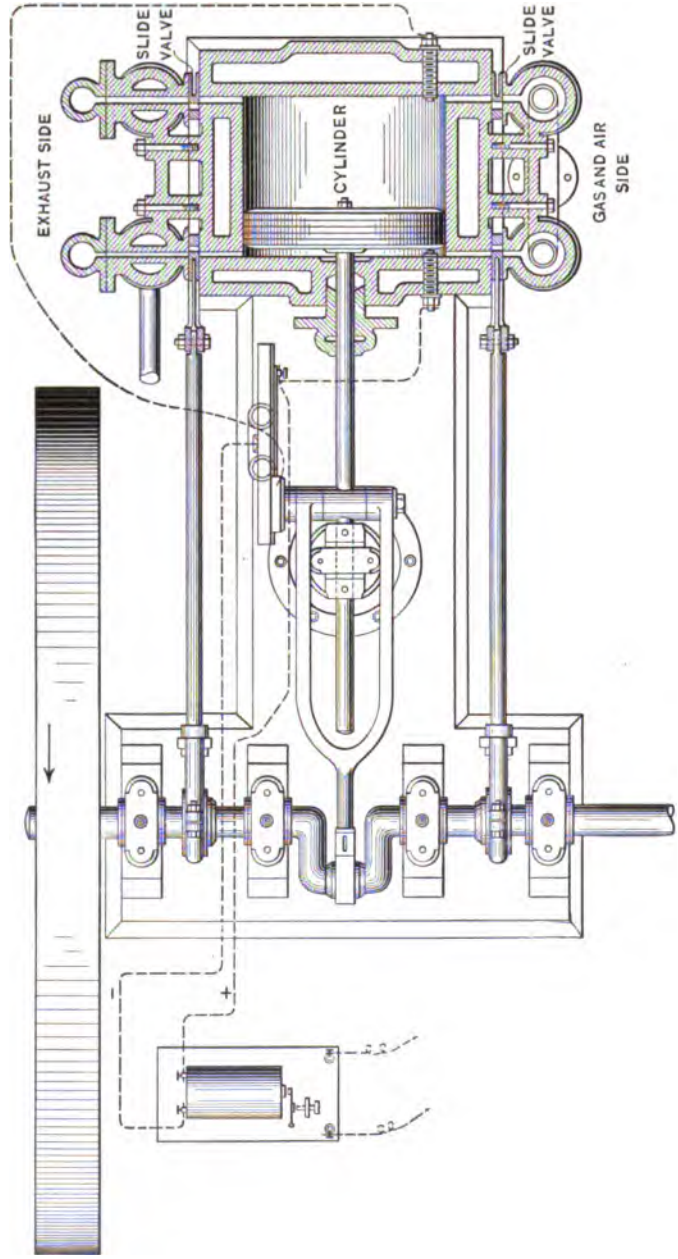


FIG. 1. — Lenoir Engine.

faction. In appearance, the engine was much like the small high-speed steam-engine of the present day. It had a cylinder, piston, crosshead, connecting-rod, crank-shaft, fly-wheel and valves moved by eccentrics. During the first year two engines were built, one of six and one of twenty horse-power. So great was the demand for these engines, that inside of five years between 300 and 400 were made in France and 100 in England. The usual reaction from undue praise and indiscriminate adoption soon came, however, when it was found that the much heralded efficiency of the engine was a myth. Tests showed that the gas consumption was a little more than 100 cubic feet of Paris gas per horse-power per hour, corresponding to an efficiency of 4.25 per cent.

The Lenoir engine shown in Fig. 1 was double-acting and the admission and exhaust were controlled by two slide-valves, one on either side of the cylinder. These slide-valves were worked by eccentrics on the main-shaft.

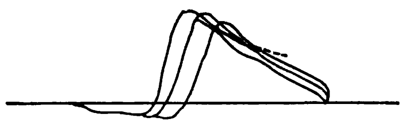


FIG. 2.—Indicator Card. Lenoir Engine.

The action of the engine was as follows: The exhaust valves being closed when the piston is at the extreme end of the stroke, as shown in the figure, the energy of the fly-wheel is sufficient to carry it forward. The air port is already slightly open, the gas valve now opens and the charge is mixed in the main part of the right-hand valve before being drawn into the cylinder by the forward stroke of the piston. Meanwhile the pressure on the other side of the piston has been reduced to that of the atmosphere. Before the admission valve is completely closed the electric spark fires the mixture and the piston is thus propelled forward to the end of the stroke, the pressure rising to five or six atmospheres. The exhaust valve has a slight lead and opens just before the end of the stroke, allowing the gases of combustion to escape at a pressure of 7 to 10 pounds above the atmosphere.

The real reasons for the uneconomical working of the Lenoir engine and others were lack of compression, incomplete expansion and heat lost through the walls. Pressures in the cylinder were

always low and difficult to obtain, and this showed that pressure generated by the gas when exploded at atmospheric pressure was insufficient. Time was also lost in obtaining an explosion as heat was applied too late and was speedily dissipated. These points are illustrated in Fig. 2, which is a pressure card from the Lenoir engine.

M. Beau de Rochas, a French engineer, was the first to formulate a complete theory of the cycles of operations which ought to be carried out in a gas-engine, in order to utilize more completely the heat supplied. Four conditions which he laid down as being essential to efficiency were as follows:

I. The largest cylinder volume, with smallest exposed surface.

II. Maximum piston speed.

III. Highest possible pressure at beginning of expansion.

IV. Greatest possible expansion.

These working conditions are now generally admitted to be necessary in order to get good economy, and are put in practice wherever gas-engines are built. At that time, however, they were revolutionary to a great extent, and it was a long time before they were adopted in actual practice.

To obtain the results which he laid down as being necessary to good efficiency, Beau de Rochas proposed to use a single cylinder and carry out the cycle in four strokes as follows:

I. Draw in charge of air and gas.

II. Compression of air and gas.

III. Ignition at dead point with subsequent explosion and expansion.

IV. Discharge of the products of combustion from the cylinder.

This cycle, known as the four-stroke or **Otto cycle,** proposed by Beau de Rochas in 1862, is now used chiefly in gas-engines.

Otto and Langen. — Four years after Beau de Rochas secured his descriptive patent in France in 1862, two men, **Otto** and **Langen,** secured a patent on an engine which embodied nothing particularly new as it was very similar to the Barsanti and Mateucci engine. These engines had vertical cylinders with a free piston which carried a rack. The explosion drove the piston upward, the rack turning a gear freely on the shaft. When the piston descended, due to its own weight and to the partial vacuum formed by the cooling of the gases underneath, this gear

drove the shaft through a ratchet or a clutch. When the piston neared the bottom of the stroke, it was cushioned slightly by compression. At this point a second clutch on a lay shaft became engaged and started the piston upward again, creating a partial vacuum; the mixture of gas and air was admitted through a slide-valve and immediately fired, driving the piston upward. This engine was more economical than Lenoir's as it used only 29 to 49 cubic feet of gas per horse-power hour, depending on the size of the unit. About 5000 engines were constructed in ten years and they were in great demand in England and Germany, though never popular in France.

Otto and Langen formed their business into a company, at Deutz near Cologne, the firm being known as the "Gas-Motoren Fabrik Deutz" — at the present time one of the largest and most successful builders of gas-engines in the world. For more than ten years these two men worked incessantly to improve their engine, but after introducing several modifications, they abandoned entirely the use of the free piston.

At the Paris Exhibition of 1878 they brought out the celebrated Otto engine which rapidly superseded their former and all other motors, and created a revolution in the construction of gas-engines. In this engine the charge of gas and air was compressed before ignition — one of the points covered by Beau de Rochas. In fact the whole cycle was carried out, according to the patent of de Rochas, in one cylinder. The cycle was divided into four piston strokes, two forward and two backward, covering two revolutions, thus obtaining one explosion and working stroke for each two revolutions in a single-cylinder, single-acting engine.

This engine, illustrated in Fig. 3, is not unlike our present single-cylinder engines in general appearance. The chief difference is in the valve. The original Otto engine was equipped with a slide-valve which controlled both the admission and ignition. This valve which moved at right angles to the axis of the engine instead of parallel to it is illustrated in Fig. 4. L is the gas and M the air admission passage. During the suction stroke Q is in communication with port a leading to the cylinder A, M registers with o and n with L. Thus gas and air are drawn through the valve and mixed before entering the cylinder. During the compression stroke the valve moves to the left until the

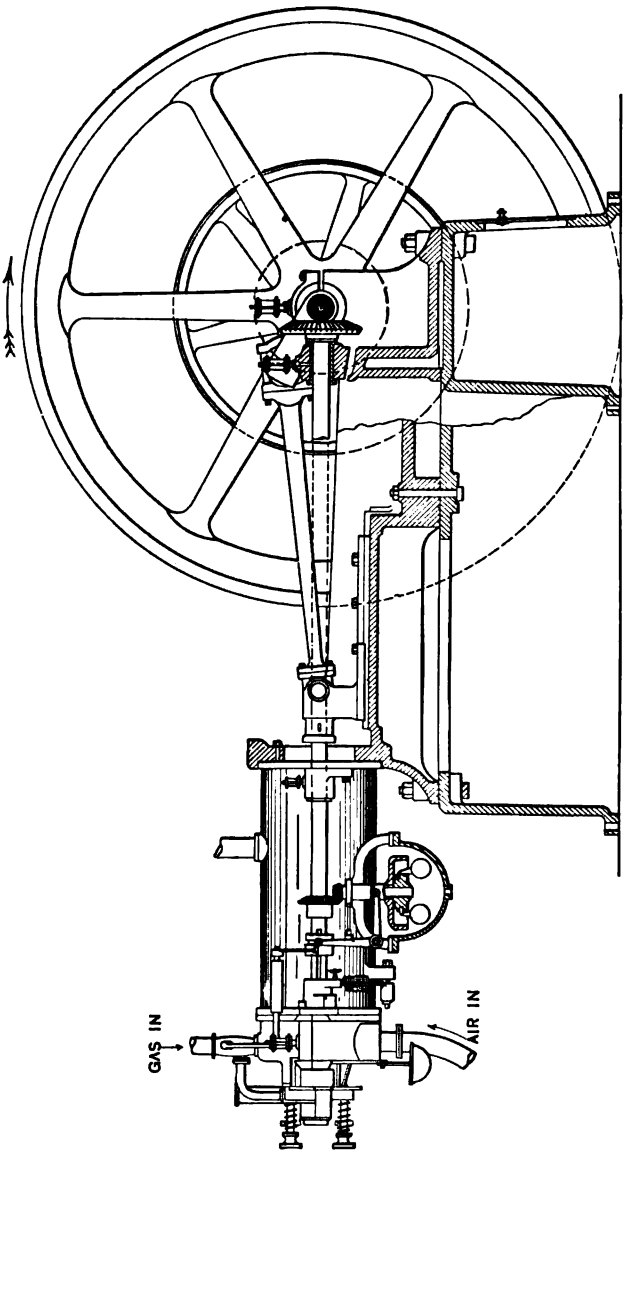


Fig. 3.—Early Otto Engine.

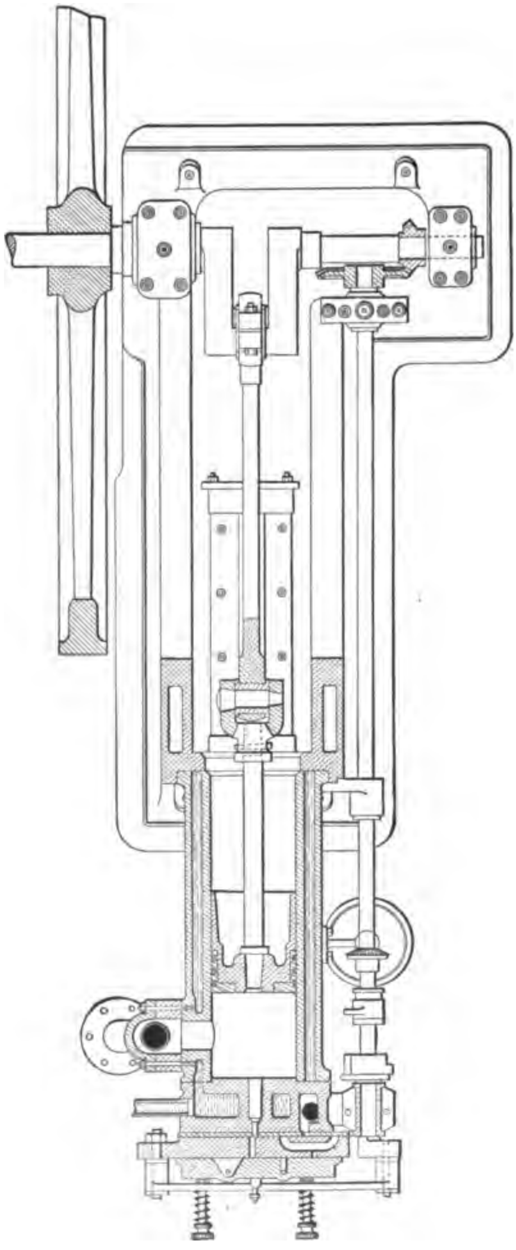


FIG. 3a. — Horizontal Section. Otto Engine.

port *N* registers with *d* which is a gas pipe. As the valve moves past the flame *B* on its return, this gas is ignited. When *N* communicates with *a*, which is at the point of ignition, the charge in the cylinder is fired.

The exhaust-valve was of the poppet type so common today. This valve was operated through a lever by a cam on the lay-shaft. The engine was governed by the "hit-and-miss" principle. A tappet worked by a cam on the lay-shaft opened a valve in the gas line on every admission stroke. When speed increased, the governor drew the tappet away from the cam so

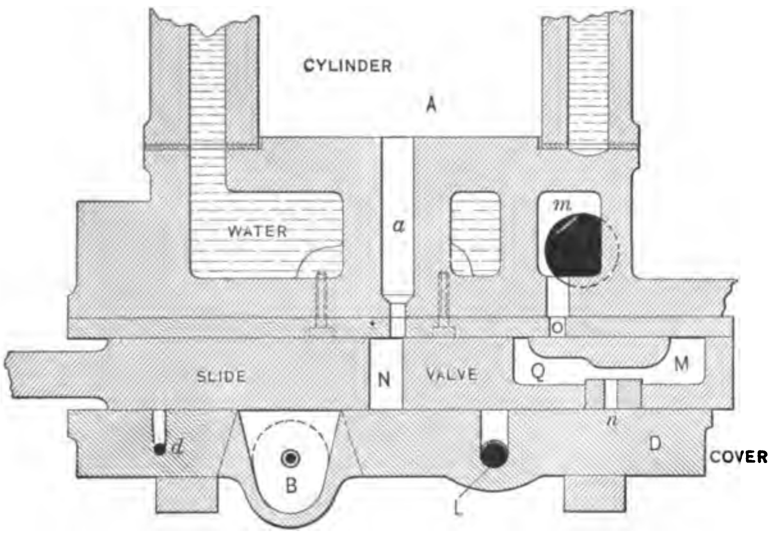


Fig. 4. — Detail of Valve. Otto Engine.

that the gas-valve remained closed and air alone was drawn into the engine on that suction stroke.

More than 30,000 of these engines were sold during the first ten years and the German firm alone sold 45,000 engines with a total of about 200,000 horse-power during the seventeen years from 1878 to 1895.

Clerk. — In order to do away with the disadvantage of having a working stroke every other revolution, **Dugald Clerk,** an English engineer, brought out a **two-stroke-cycle** engine which gave an explosion every revolution. The work of admitting a new charge was given over to an auxiliary cylinder, called the dis-

placer. As the main piston approached the end of the expansion, exhaust ports were uncovered and pressure was immediately reduced to that of the atmosphere. At this point the displacing cylinder forced the new charge in at the head of the motor cylinder, driving out the burned gases. Ignition was accomplished by means of an ignition valve somewhat similar to the one used on the Otto engine. The practical difficulties which were presented in the manufacture of this engine were so great that attention was again turned towards the development of the four-cycle engine, and it was not until about ten years later that the two-stroke operation was commercially successful.

Brayton. — Before Otto took out his patent for the engine which influenced practically all gas-engines built afterwards, an American, **George B. Brayton,** took out patents on a gas- and an oil-engine. Both these engines had one feature that was unique, in that combustion took place at constant pressure. The engines were of the walking-beam type, and connected to the beam, so as to give a shorter stroke than the working cylinder, was an air cylinder. The gas and air were compressed in this cylinder and forced into the working cylinder during the outstroke. As the mixture entered the working cylinder, it passed over a flame which ignited it so that it burned at constant pressure. A wire gauze prevented the flame from striking back into the passages and compression cylinder. The gas-engine was never successful so oil was resorted to for fuel. In the oil-engines air alone was compressed into a receiver. As the air was admitted to the working cylinder it passed over a sponge saturated with kerosene or gasoline, then over a flame which burned the mixed air and vapor at constant pressure as in the gas-engine. In both of these engines the inlet-valve closed at about midstroke and expansion took place from there to the end of the stroke. These engines were built in large numbers and possessed distinct advantages over any previcusly made.

CHAPTER III

LAWS OF GASES — THERMAL LINES — PROCESSES — ENTROPY

While this book is written with the general understanding on the part of the author that the student already has a knowledge of thermodynamics, it will not be out of place at this point to review some of the more important laws of gases that will be used later on in the development of this work.

Boyle's Law. — At constant temperature the volume of a given weight of gas varies inversely as the pressure.

Charles' Law. — With the volume constant the change of pressure of a gas is proportional to the change of temperature.

By a combination of these two experimental laws the characteristic equation $PV = MBT$ may be deduced. P is the pressure of the gas under consideration in pounds per square foot, V is the volume in cubic feet, M, the weight of the gas in pounds, T, the absolute temperature in degrees Fahrenheit and B is a constant which may be found as follows:

$$B = \frac{PV}{T}, \text{ when } M = 1.$$

Substituting the known values for air at certain conditions of temperature and pressure, we have

$$B = \frac{2116.32}{0.08071 \times 491.6} = 53.34,$$

2116.32 = pressure per square foot with barometer at 760 millimeters,

$$V = \frac{1}{\text{wt. of 1 cu. ft. air}} = \frac{1}{0.08071},$$

$$T = 491.6° \text{ F.}$$

The value of B for any other gas may be found by using the weight per cubic foot of that gas in place of 0.08071. Thus for hydrogen

$$B = \frac{2116.32}{0.00559 \times 491.6} = 767.37.$$

For a **mixture** of gases the value of B_{mix} may be found by multiplying the fractional weight of each constituent by the value of B for that gas and adding the products. Thus for a producer gas with the following composition by weight:

$$
\begin{aligned}
H_2 &= 0.60 \text{ per cent} \\
CO &= 23.08 \\
CH_4 &= 1.44 \\
CO_2 &= 10.88 \\
N_2 &= \underline{64.00} \\
& 100.00 \text{ per cent}
\end{aligned}
$$

we have

$$
\begin{aligned}
0.006 \times 767.37 &= 4.604 \\
0.2308 \times 55.142 &= 12.727 \\
0.0144 \times 96.314 &= 1.387 \\
0.1088 \times 35.09 &= 3.820 \\
0.64 \times 54.985 &= \underline{35.190} \\
& 57.748 \\
B_{mix} &= 57.748.
\end{aligned}
$$

The application of the general equation $PV = MBT$ may best be shown by the use of examples.

What will be the size of the gas holder required to contain 1200 pounds of the producer gas mentioned above, when the temperature is 82 degrees, gauge pressure 5 ounces per square inch?

$$V = \frac{MBT}{P},$$

$$
\begin{aligned}
P &= 2116.32 + \tfrac{5}{16} \times 144 = 2161.32, \\
M &= 1200, \\
B &= 57.748, \\
T &= 459.6 + 82 = 541.6,
\end{aligned}
$$

$$V = \frac{1200 \times 57.748 \times 541.6}{2161.32} = 17{,}364 \text{ cubic feet.}$$

Characteristic Surface.— The characteristic equation $f(P, V, T) = 0$, having three variables, may be represented by a surface. A state or condition of a substance is defined by its coördinates P_1, V_1, T_1, and this condition is therefore represented by a point on the surface. If the condition changes, a second point with coördinates P_2, V_2, T_2 will represent the new condition. The succession of points between the initial and final conditions will be

represented by a succession of points on the surface. The surface representing the equation $PV = BT$ is shown in Fig. 5.

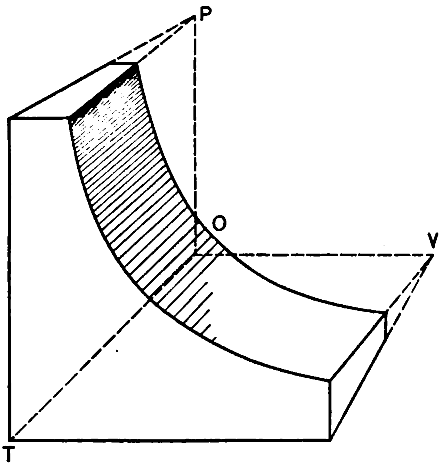


Fig. 5. — Characteristic Surface.

Thermal Lines. — If, during a change of condition, the **temperature** of a substance shall remain **constant,** the point will evidently move on the characteristic surface parallel to the PV-plane. Such a change of condition is called an **isothermal,** and the curve described by the point is the isothermal curve. By taking different constant values for the temperature, we get a complete representation of the characteristic equation. For the perfect gas the isothermal lines consist of a system of **equilateral hyperbolas** having the general equation

$$PV = \text{constant}.$$

If the **pressure** remains constant during a change of condition, the point will move parallel to the TV-plane and the projection of the path on the PV-plane will be a *straight* line parallel to OV, as AB in Fig. 6.

If the substance changes its condition at **constant volume,** the point moves parallel to the PT-plane and the projection of the path on the PV-plane is a **straight line** parallel to the P-axis as CD, Fig. 6.

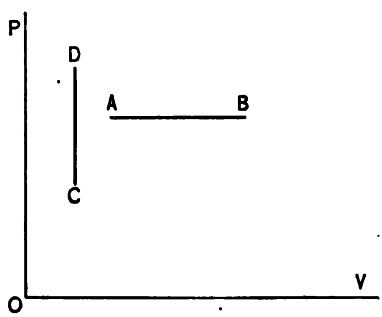


Fig. 6. — Constant Pressure and Volume Lines on PV-Plane.

Besides the cases just given, there are others which are not of prime importance in gas-engine work. One, however, is of great importance and will be taken up later on. This is the **adiabatic,** representing a change of condition in which the system neither receives nor gives out heat.

Specific Heat. — In general, the heat required to raise the temperature of a body 1 degree under given external conditions is called the **thermal capacity** of the body. Thus the thermal capacity of 1 gram of water at 17.5° C. is 1 calorie, and that of 1 pound of water at 63.5° F. is 1 British thermal unit.

The **specific heat** of a substance at a given temperature is the ratio of the thermal capacity of that substance at this temperature to the thermal capacity of an equal mass of water at some chosen standard temperature. If we take 17.5° C. (63.5° F.) as the standard temperature, and denote by L the thermal capacity per unit weight, then the specific heat c is given by the relation

$$c = \frac{L_t \text{ (of substance)}}{L_{17.5} \text{ (of water)}}.$$

For water $L_{17.5} = 1$ calorie. It follows, then, that the specific heat at the temperature t is numerically equal to the thermal capacity of a unit weight of the substance at the same temperature; thus, at 100° C. the thermal capacity of 1 gram of water is found to be 1.005 calories, and the specific heat is

$$\frac{L_{100}}{L_{17.5}} = \frac{1.005 \text{ cal.}}{1 \text{ cal.}} = 1.005.$$

Adiabatic Processes. — When a system, in changing its condition, has no thermal communication with other bodies, and, therefore, neither absorbs nor gives out heat, the change of condition is said to be **adiabatic.** In general, adiabatic changes are possible only when the system is enclosed in a non-conducting envelope. Rapid changes of condition are approximately adiabatic since time is required for conduction or radiation of heat; thus, the compression of air in an " air-cooled " compressor cylinder is practically adiabatic as the time is so short that little heat enters the cylinder walls. Since there is no heat added during adiabatic compression nor taken away during adiabatic expansion, the **work** done on the system during compression or by the system during expansion is at the expense of the **intrinsic energy.**

The projection on the PV-plane of the path of a point during an adiabatic change of condition gives the **adiabatic** curve. This curve is somewhat steeper than the isothermal curve as is shown by Fig. 7, where AB is the adiabatic and CD the isothermal. This follows from the fact that during adiabatic expansion the energy

decreases and as a result the temperature falls; hence for the same final volume the temperature, and therefore the pressure, is lower for the adiabatic expansion than for the isothermal expansion.

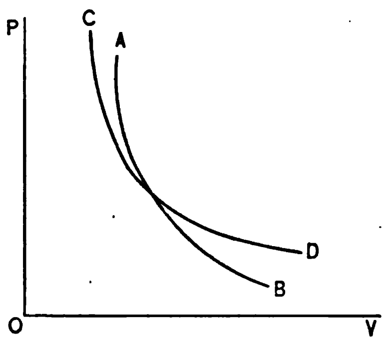


FIG. 7. Adiabatic and Isothermal Curves on PV-Plane.

Availability of Energy. — Either **mechanical** or **electrical** energy can be completely transformed into the other in theory and nearly so in practice. Either one can be completely transformed into **heat.** Examples of these changes are the dynamo driven by a steam- or gas-engine for the change from mechanical to electrical, and the prony brake used for absorbing the power of an engine or the rheostat used for absorbing the power of a dynamo for the change from mechanical and electrical to heat energy. On the other hand experience shows that heat energy is not capable of complete conversion into mechanical work, and to get even a part of heat energy transformed into mechanical energy, certain conditions must be satisfied. As a first condition there must be two bodies of different temperatures; it is impossible to derive work from the heat of a body unless there is available a second body of lower temperature. Even if we have two bodies of different temperatures and place them in contact, the heat will flow from the hotter to the colder and no work will be obtained. Hence, as a second condition, the systems or bodies of different temperatures must be kept apart and a third system used to convey the energy. This third system may be called the medium. In the steam plant, for instance, the boiler furnace is the system of high temperature, the condenser is the system of low temperature and the steam is the medium. This medium is placed in contact with the hot system and receives heat from it; then by a change in condition expansion gives up energy in the form of work and delivers to the system of low temperature an amount of heat smaller than the amount taken from the source of high temperature, the difference being transformed into work.

If the total heat taken from the furnace be called Q_1 and the part

rejected into the condenser Q_2, then the heat transformed into work will be $Q_1 - Q_2$. This difference is also the available part of Q_1, and the part that is rejected into the condenser is the unavailable part or the waste. The ratio $\frac{Q_1 - Q_2}{Q_1}$ is called the availability of Q_1 for the transformation into mechanical energy. In general the term signifies the fraction of the energy of a given system in a given state or condition that can be transformed into mechanical work.

Reversibility. — A process is said to be reversible when the following conditions are fulfilled:

1. When the direction of the process is reversed, the system taking part in the process can assume in inverse order the conditions traversed in the direct process.
2. The external actions are the same for the direct and reversed processes or differ by an infinitesimal amount only.
3. Not only the system undergoing the change but all the connected systems can be restored to initial conditions.

A process that fails to meet these requirements in any particular is an irreversible process.

Suppose a cylinder confines gas to act on a piston, as in a steam- or gas-engine. See Fig. 8. If A is the piston area, the pressure acting on the face of the piston is pA, and for equilibrium this pressure must be equal to the force F. If we now assume that the force pA is slightly greater than F, the piston will move slowly to the right and the confined gas will assume a succession of conditions indicated by the curve AB. If at the condition B the motion is arrested and F is made infinitesimally greater than pA for all positions of the piston, the series of conditions from B to A will be retraced and the system (the confined gas in this case) will be brought back to its original state without leaving changes in outside bodies. The reverse process is accomplished by an infinitely small modification of the external force F. Therefore the process is reversible.

As examples of **irreversible** processes we might look at the following:

1. **Friction,** as in a **journal.** When a journal turns in a bearing, friction is always present and the work of friction is transformed into heat which is radiated. It would be folly, however, to try to turn the journal by heating the bearing. Thus the process is irreversible.

2. **Conduction in partitions.** We have all experienced the flow of heat from a warm room into the colder air but no one has yet been fortunate enough to experience the flow of heat from a cold atmosphere into a warm room. Hence the irreversibility.

Second Law of Thermodynamics. — According to the **First Law of Thermodynamics** the total quantity of energy in a system of bodies cannot be increased nor decreased by any change, reversible or irreversible, that may occur within the system. The

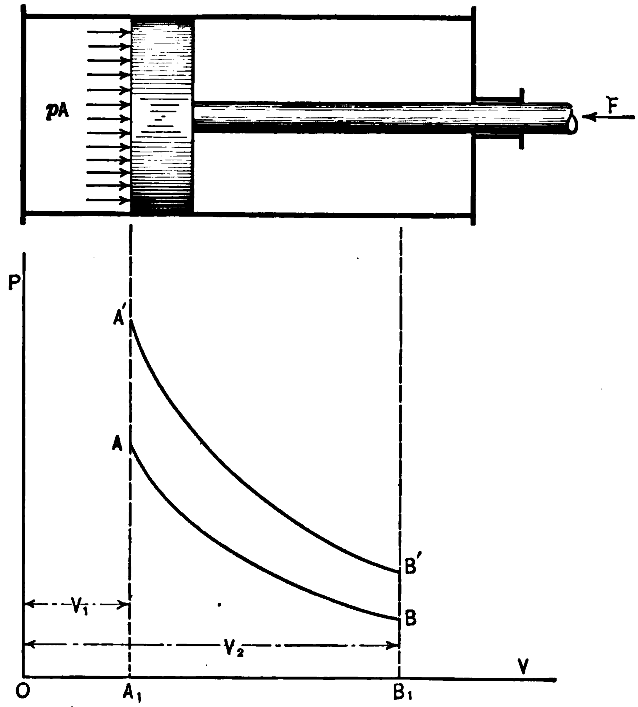


Fig. 8.

total energy, however, is not of as great importance as the available energy, and experience shows that a change within the system usually results in a change in the availability of the energy of the system.

Professor Goodenough, in his "Principles of Thermodynamics," deduces the following general laws to explain the **Second Law of Thermodynamics** and the **degradation of energy**:

1. No change in a system of bodies that can take place of itself can increase the available energy of the system.

2. An irreversible change causes a loss of availability.
3. A reversible change does not affect the availability.

Carnot's Cycle. — Keeping in mind the statements of the preceding article we may now look at the cycle of processes first described by Carnot in 1824. A cycle has been defined as a series of operations at the end of which the working substance reaches initial conditions. The Carnot cycle represents the best that can be done in the conversion of heat energy into mechanical energy. It cannot be used in an actual engine but is of theoretical interest as a criterion of the best results obtainable. The apparatus for this cycle consists of a source of heat S, at a temperature T; a refrigerator at a temperature T_2; and a working fluid or medium M.

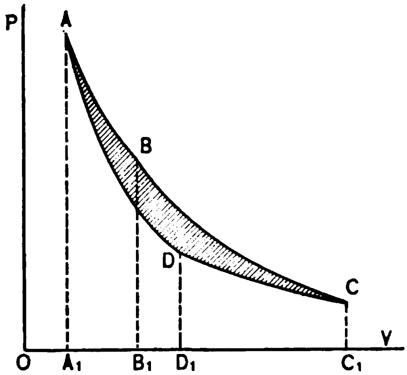


FIG. 9.

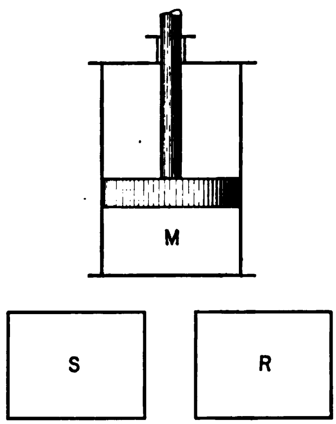


FIG. 10.

Assume the medium to be initially at a condition shown by B, Fig. 9, at the temperature T_1 of the reservoir S, then expand adiabatically until its temperature falls to T_2, the temperature of the body R. By this expansion the second condition C is reached and the work done by the medium is represented by BCC_1B_1. The expansion is assumed to proceed slowly so that the pressures on the two faces of the piston are equal, and the process is therefore reversible. The cylinder is now placed in contact with R so that heat can flow from M to R and the medium is compressed. The work represented by C_1CDD_1 is done on the medium, and the heat Q_2 passes from the medium to the refrigerator. The process is

again assumed to be so slow as to be reversible. From the condition D the medium is now **compressed adiabatically,** the cylinder being removed from R until its temperature again becomes T_1, that of the source S. During the third process work represented by D_1DAA_1 is done on the fluid. Finally the cylinder is placed in contact with S and the fluid allowed to expand at the constant temperature T_1 to the initial condition B. Work represented by the area A_1ABB_1 is done by the fluid during the process, and the temperature is kept constant by the flow of heat Q_1 from S to M.

The area $ABCD$ enclosed by the four curves of the cycle represents the **mechanical work** gained; that is, the excess of the work done by the medium over the work done on the medium. Denoting this by W we have $Q_1 - Q_2 = \frac{W}{J}$, where J is Joule's equivalent.

The **efficiency** of the cycle is $\frac{Q_1 - Q_2}{Q_1}$ and is equal to $\frac{T_1 - T_2}{T_1}$.

Since all processes of the Carnot cycle are reversible, it is evident that they may be traversed in reverse order. Carnot set forth the principle that of all engines working between the same source and the same refrigerator, no engine can have a greater efficiency than that of the reversible engine, and thus the efficiency is independent of the working fluid.

Available Energy and Waste. — **Carnot's cycle** gives us a means of measuring the **available energy** of a system and the waste due to an irreversible change of condition. Suppose a quantity of heat ΔQ is absorbed by the system at a temperature T, and we wish to find the part of this heat that can be transformed into work. No device can transform a larger portion of heat ΔQ into work than the Carnot engine. If T_0 is the lowest temperature that can be obtained for a refrigerator, the fraction $\frac{T - T_0}{T}$ of Q can be transformed into work by the Carnot engine and this may be called the available heat, or the available part of ΔQ is $\Delta Q \left(1 - \frac{T_0}{T}\right)$; the unavailable part is $\Delta Q - \Delta Q\left(1 - \frac{T_0}{T}\right) = \Delta Q \frac{T_0}{T}$.

For conduction of a quantity of heat Q from a temperature of T_1 to a temperature of T_2, it may be shown that the unavailable energy is increased by an amount $T_0\left(\frac{Q}{T_2} - \frac{Q}{T_1}\right)$.

During an irreversible conversion of work into heat through the agency of friction, the amount of heat thus generated we may call ΔQ. $\frac{T - T_0}{T} \Delta Q = \left(1 - \frac{T_0}{T}\right) \Delta Q$ may be transformed back into work. Of the work $J \Delta Q$ expended in producing heat ΔQ, the part $J \Delta Q \left(1 - \frac{T_0}{T}\right)$ may be recovered in the form of work. The remainder, $J T_0 \frac{\Delta Q}{T}$ is rendered unavailable.

Entropy. — All the expressions for the increase in unavailable energy derived above have a common factor T_0. From this it appears that the available energy varies with the temperature of the coldest refrigerator we may have and that the lower T_0, the less the unavailable heat.

The other factor has the form $\frac{Q}{T}$, or in some cases $\int \frac{dQ}{T}$. To this second factor, which, when multiplied by T_0 gives as a product the increase of unavailable energy, the name increase of entropy is given. The change of entropy of a system is therefore a measure of the change of unavailable energy of the system; an increase of entropy involves an increase of unavailable energy and vice versa. Entropy may then be defined as follows:

If, from any cause whatever, the unavailable energy of a system is increased and the increase is divided by T_0, the lowest temperature available for a cold body, the quotient is the increase of entropy of the system.

The "system" spoken of may be either a single substance, as the medium employed in a heat motor, or it may be all the bodies taking part in the process. When we apply the idea of increase of entropy to the system composed of all the bodies involved in the process, we are led to the conception that the increase of entropy measures the degradation of energy of the process. If this notion is combined with that expressed in the second law, we arrive at the following principles:

1. Any process that can proceed of itself is accompanied by an increase of the entropy of the system of bodies involved in the process.

2. The direction of a process, physical or chemical, that occurs of itself is such as will bring about an increase of entropy in the system.

Another definition of entropy is that derived by considering only one of the bodies taking part in a process, say the medium M in the Carnot cycle.

The change of entropy of a system corresponding to a change of the system from condition 1 to condition 2 is the definite integral $\int_{T_1}^{T_2} \frac{dQ}{T}$ taken along any reversible path between the two conditions.

The entropy thus defined is purely relative. We are never concerned with the absolute value of the entropy of a system in a given condition; what is desired is the change of entropy corresponding to a given change of condition. For the convenience of calculation we may assume the zero of entropy to be the entropy of a system in some specified state.

Temperature-entropy Plane. — Since the entropy of a system measured from an arbitrary zero is dependent only on the condition of the system, that is, the entropy is a function of the coördinates of the system, entropy may thus be included in the coördinates used to define a system and used with the other coördinates to form pairs. For convenience we use it with temperature only, giving us, in addition to the PV-plane, the TS-plane. The other planes formed by combinations of the other coördinates are not of especial importance in gas-engine design.

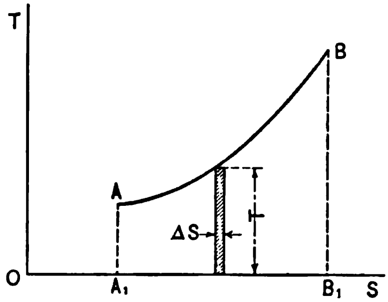


FIG. 11.

From the second definition of entropy we have

$$S_1 - S_2 = \int_{T_2}^{T_1} \frac{dQ}{T}, \tag{1}$$

or

$$dS = \frac{dQ}{T}, \tag{2}$$

and

$$T\,dS = dQ. \tag{3}$$

Let the curve AB, Fig. 11, be the path of a point projected on the TS-plane, the points A and B representing respectively the initial and final conditions. The area A_1ABB_1 represents the limit of the sums of the terms of the type $T\Delta S$, that is, the area

under the curve AB is the definite integral $\int_{S_1}^{S_2} T\,dS$. From equation (3) this is equal to the heat absorbed by the system from external sources during the change. One restriction must be made. *The change in condition must not involve any internal irreversible effects. In other words, the area under AB is equal to the heat absorbed only when AB is reversible.*

EXERCISES

1. What would be the weight of 2500 cu. ft. of air at 80° F., 145.3 lbs. gauge pressure?

2. Find the value of B_{mix} for a gas of the following composition by weight:

	Per cent
H_2 =	5.00
CO =	24.00
CH_4 =	2.68
CO_2 =	12.00
N_2 =	50.00
Air =	6.32
	100.00

3. Find the height of a tank 30 ft. in diameter, to hold 1400 lbs. of the gas in Prob. 2. Pressure 6 oz. per sq. in., temperature 70° F.

CHAPTER IV

PRINCIPAL GAS-ENGINE CYCLES

Carnot — Brayton — Lenoir — Diesel — Otto

General. — Many cycles have been proposed and experimented on for internal combustion engine work but few have proved to be practical. One of historical and theoretical interest is the **Carnot** cycle mentioned in the last chapter. This cycle has never been realized in actual practice, although many experimenters have attempted to build engines to use it. The **Lenoir** cycle was first successfully used about 1860 and many engines were built to use that cycle. It proved to be low in actual efficiency and was abandoned when the **Beau de Rochas** or **Otto** cycle came into use about fifteen years afterward. The **Otto** may be used on either the two-stroke or four-stroke principle. If the four-stroke principle is used, the exhaust and suction strokes do not enter into the thermal analysis of the cycle. When the two-stroke principle is used, these events take place outside the cylinder and consequently have no bearing on the thermal analysis of the cycle. The **Brayton** cycle was introduced in the United States in 1873. This cycle was composed of a **constant-pressure** line where **heat** was **added** by the burning of the mixture, **adiabatic expansion** to the **atmospheric pressure, exhaust** or **discharge** of heat at **constant pressure,** and **adiabatic compression.** This cycle was used for a short time and discarded. The **Diesel** cycle is somewhat similar to the **Brayton** in that the heat is added at **constant pressure** before **adiabatic expansion.** Heat is **discharged,** however, at **constant volume** instead of at **constant pressure** as in the **Brayton** cycle. **Diesel** first tried to use the **Carnot** cycle but found the necessary pressures and resulting temperatures too high for practical use.

Carnot Cycle. — Referring to Fig. 9, the heat **added** during **isothermal expansion** AB is

$$Q_{ab} = \frac{1}{J} P_a V_a \log_e \frac{V_b}{V_a}, \tag{1}$$

and the heat given up or exhausted is

$$Q_{cd} = \frac{1}{J} P_c V_c \log_e \frac{V_d}{V_c} = -\frac{1}{J} P_c V_c \log_e \frac{V_c}{V_d}. \tag{2}$$

The heat transformed into work is then

$$\frac{W}{J} = Q_{ab} + Q_{cd} = \frac{1}{J}\left(P_a V_a \log_e \frac{V_b}{V_a} - P_c V_c \log_e \frac{V_c}{V_d}\right). \tag{3}$$

Since the temperature at A is T_1,

$$P_a V_a = MBT_1 \tag{4}$$

and

$$P_c V_c = MBT_2. \tag{5}$$

Since BC is an **adiabatic** we have

$$\left(\frac{V_c}{V_b}\right)^{n-1} = \frac{T_1}{T_2}, \tag{6}$$

and for the adiabatic DA,

$$\left(\frac{V_d}{V_a}\right)^{n-1} = \frac{T_1}{T_2}. \tag{7}$$

Then from (6) and (7)

$$\frac{V_c}{V_b} = \frac{V_d}{V_a} \quad \text{or} \quad \frac{V_c}{V_d} = \frac{V_b}{V_a}. \tag{8}$$

Inserting the values found in (4), (5) and (8) in (3) we get

$$\frac{W}{J} = \frac{MB}{J}(T_1 - T_2) \log_e \frac{V_b}{V_a}. \tag{9}$$

Then

$$E = \frac{\frac{W}{J}}{Q_{ab}} = \frac{MB\,(T_1 - T_2) \log_e \frac{V_b}{V_a}}{MBT_1 \log_e \frac{V_b}{V_a}} = \frac{T_1 - T_2}{T_1}. \tag{10}$$

It is important to notice here that T_1 is the **highest** and T_2 the **lowest** temperature reached in the cycle.

On the ***TS*-plane** the **Carnot** cycle will be in the form of a **rectangle,** as shown in Fig. 12. Heat is added at **constant temperature** along AB. **Adiabatic expansion** takes place from B to C, **heat** is **exhausted** along CD and **adiabatic compression** occurs from D to A.

Since the **processes** in this cycle are **reversible,** the area under the lines represents the heat added from an external source. The **heat added** along AB is then

$$(S_1 - S_2)\, T_1. \tag{1}$$

The heat rejected along CD is

$$(S_1 - S_2)\,T_2. \tag{2}$$

The **heat transformed** into work is the **difference**

$$(S_1 - S_2)\,(T_1 - T_2). \tag{3}$$

The **efficiency** is, by definition (3) divided by (1),

$$\frac{(S_1 - S_2)}{(S_1 - S_2)} \cdot \frac{(T_1 - T_2)}{T_1} = \frac{T_1 - T_2}{T_1}. \tag{4}$$

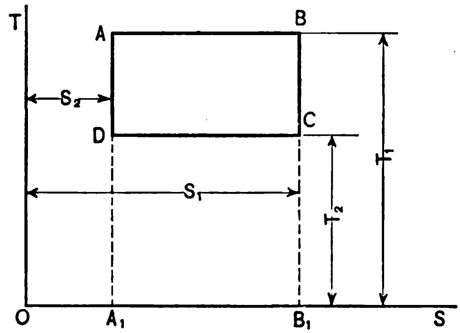


FIG. 12.

The Lenoir Cycle. — In the **Lenoir cycle combustion** is effected **without** previous compression. The charge is drawn into the cylinder during the **first** part of the stroke and exploded at a point suitable to the load. The pressure then **rises,** due to **heating** of the charge at approximately **constant volume.**

Figs. 13 and 14 represent, respectively, the PV- and TS-diagrams of this cycle.

From B to C the heat is **added** to the charge by **combustion** at **constant volume.** This heat is

$$Q_1 = c_v\,(T_c - T_b). \tag{1}$$

The heat rejected from A to B is

$$Q_2 = c_p\,(T_a - T_b). \tag{2}$$

The **efficiency,** being the heat transformed into work divided by the heat supplied, is

$$E = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{c_p\,(T_a - T_b)}{c_v\,(T_c - T_b)} = 1 - n\frac{\frac{T_a}{T_b} - 1}{\frac{T_c}{T_b} - 1} \tag{3}$$

since $\frac{c_p}{c_v} = n$. During the period of **constant pressure** from A to B

$$\frac{T_a}{T_b} = \frac{V_a}{V_b}, \tag{4}$$

and during the period of **constant volume** from B to C

$$\frac{T_c}{T_b} = \frac{P_c}{P_b}. \tag{5}$$

Also from the relations of **pressure** and **volume** along an **adiabatic**

$$\frac{P_c}{P_a} = \left(\frac{V_a}{V_c}\right)^n. \tag{6}$$

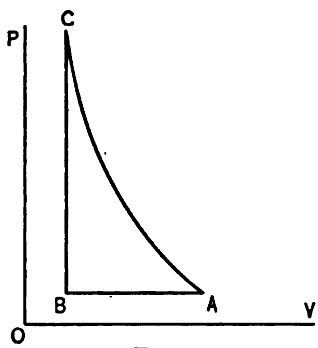


FIG. 13.

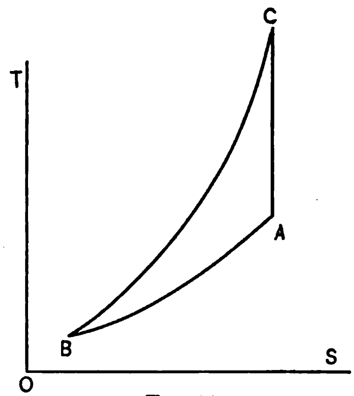


FIG. 14.

Lenoir Diagram.

But as $P_a = P_b$ and $V_c = V_b$,

$$\frac{P_c}{P_b} = \frac{T_c}{T_b} = \left(\frac{V_a}{V_b}\right)^n. \tag{7}$$

Let $\frac{V_a}{V_b} = r$, the **ratio** of **expansion**;

then
$$\frac{T_c}{T_b} = r^n, \tag{8}$$

and the efficiency is

$$E = 1 - n\frac{r - 1}{r^n - 1}. \tag{9}$$

By transformation of equation (1) we get

$$\frac{T_c}{T_b} = \frac{Q_1}{c_v T_b} + 1 = r^n, \tag{10}$$

for the **range** of **temperature** during the cycle.

The **maximum pressure** reached will be

$$P_c = r^n P_b = P_b\left(\frac{Q_1}{c_v T_b} + 1\right). \tag{11}$$

The Brayton Cycle. — In the **Brayton engine** the mixture of air and gas was compressed into a reservoir to a pressure of about 60 pounds per square inch and from this reservoir it flowed into the working cylinder, where it was **ignited** by a **flame.** A wire gauze **diaphragm** was used to prevent the **flame flashing back** into the reservoir. The mixture was thus burned in the working cylinder, **without explosion,** during about one-half the stroke of the piston. By proper regulation of the admission valve the rate of combustion was regulated to give practically **constant pressure** during **combustion.**

The ideal cycle of operations is as shown in Fig. 15.

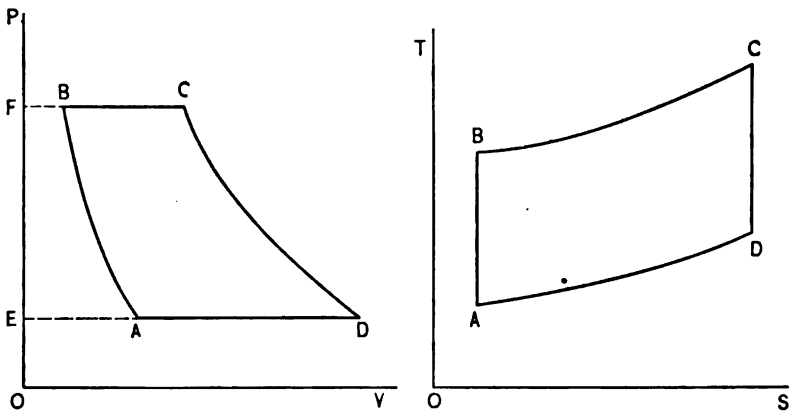


FIG. 15. FIG. 16.

Brayton Diagram.

From E to A the charge is drawn into the **compression** cylinder.

From A to B **adiabatic compression** occurs.

From B to C **expulsion** from compressor occurs **simultaneously** with **admission** and **combustion** in motor cylinder.

From C to D **adiabatic expansion** takes place after **cut-off.**

From D to E **burned gases** are **expelled.**

The area $EABF$ represents the negative work of the **compressor** and the area $EDCF$ the **work obtained** from the **motor;** hence, the area $ADCB$ represents the **net available work.**

Heat generated $Q_{bc} = Mc_p (T_c - T_b)$. (1)

Heat discharged $Q_{da} = Mc_p (T_d - T_a)$. (2)

Work done in form of heat energy $= Q_{bc} - Q_{da}$

$$= Mc_p (T_c - T_b - T_d + T_a). \quad (3)$$

$$E = \frac{Mc_p (T_c - T_b - T_d + T_a)}{Mc_p (T_c - T_b)} = 1 - \frac{T_d - T_a}{T_c - T_b} = 1 - \frac{T_a}{T_b}\frac{\dfrac{T_d}{T_a} - 1}{\dfrac{T_c}{T_b} - 1}. \quad (4)$$

Since the pressures are constant, we have

$$\frac{T_c}{T_b} = \frac{V_c}{V_b} \quad \text{and} \quad \frac{T_d}{T_a} = \frac{V_d}{V_a}.$$

Also from the adiabatic relation we have

$$\frac{T_a}{T_b} = \left(\frac{V_b}{V_a}\right)^{n-1},$$

$$P_b = \left(\frac{V_a}{V_b}\right)^n P_a,$$

$$P_c = \left(\frac{V_d}{V_c}\right)^n P_d.$$

Since $P_b = P_c$ and $P_a = P_d$,

therefore $\dfrac{V_a}{V_b} = \dfrac{V_d}{V_c}$,

and $\dfrac{V_c}{V_b} = \dfrac{V_d}{V_a} = \dfrac{T_c}{T_b} = \dfrac{T_d}{T_a}$.

Hence $E = 1 - \dfrac{T_a}{T_b} = 1 - \left(\dfrac{V_b}{V_a}\right)^{n-1} = 1 - \dfrac{1}{r^{n-1}}$, (5)

where $r = \dfrac{V_a}{V_b}$.

The **temperature range** may be found as follows:

$$\frac{T_c}{T_b} = \frac{Q_{bc}}{c_p T_b} + 1. \quad \text{[(from Eq. (1)]}$$

$$\frac{T_b}{T_a} = \left(\frac{V_a}{V_b}\right)^{n-1} = r^{n-1},$$

or $T_b = T_a r^{n-1}$;

hence $\dfrac{T_c}{T_a} = r^{n-1}\left(\dfrac{Q_{bc}}{c_p T_b} + 1\right)$. (6)

The **maximum pressure** will be

$$P_c = P_a r^n. \quad (7)$$

The *TS*-diagram of the ideal **Brayton** cycle is as shown **in** Fig. 16. It will be noticed later that the appearance is very much the same as the *TS*-diagram of the **Otto** cycle, the difference being that the lines connecting the **adiabatics** in the **Brayton** cycle are **constant-pressure** lines while in the **Otto** they are **constant-volume** lines.

The Diesel Cycle. — In the **Diesel** engine **air** without fuel is **compressed** to a pressure of about 500 pounds per square inch. The **temperature** at the end of compression is consequently higher than the **ignition** temperature of the fuel. At the end of the compression stroke fuel is injected into the compressed air and burns immediately. By proper **regulation** of the **fuel valve**

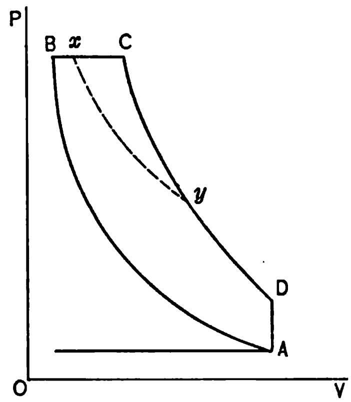


FIG. 17.

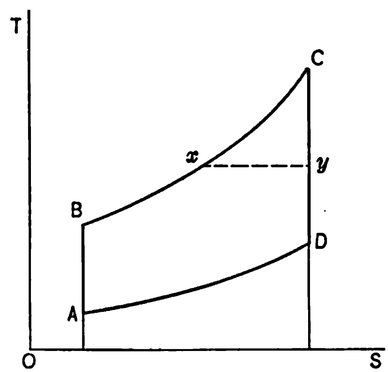


FIG. 18.

Diesel Diagram.

the air may be made to expand for a short period at **constant pressure,** or, if desired, at a **falling pressure** and approximately **constant temperature.** As in the **Brayton** engine **governing** is effected by cutting off the fuel supply earlier or later.

The ***PV*-diagram** of the ideal **Diesel** cycle is shown in Fig. 17. It was the **original** aim of **Diesel** to regulate the fuel supply so that a short period of **combustion** *Bx* would be followed by **isothermal expansion** *xy*. He found, however, that in order to get this result it would be necessary to compress the air to a **higher pressure** than could be taken care of in **practice.** The **mechanical difficulties** were so great that they could not be overcome in an engine that would be practical. The cycle that was finally brought out by the late Dr. Diesel is *ABCD.*

On the **TS-plane** this cycle appears as in Fig. 18. BC is a **constant-pressure** curve, CD **adiabatic expansion,** DA **constant volume** and AB **adiabatic compression.**

The **heat added** from B to C is

$$Q_{bc} = Mc_p (T_c - T_b). \tag{1}$$

The **heat rejected** from D to A is

$$Q_{da} = Mc_v (T_d - T_a). \tag{2}$$

Heat **transformed into work** is

$$Q_{bc} - Q_{da} = M [c_p (T_c - T_b) - c_v (T_d - T_a)]. \tag{3}$$

$$E = \frac{c_p (T_c - T_b) - c_v (T_d - T_a)}{c_p (T_c - T_b)} = 1 - \frac{1}{n}\frac{(T_d - T_a)}{(T_c - T_b)}, \tag{4}$$

where $n = \frac{c_p}{c_v}.$

The Otto Cycle.—This cycle, which is of utmost importance on account of its wide adoption for gas- and oil-engines, is shown on the PV-plane in Fig. 19.

From E to A the **explosive mixture** is drawn into the cylinder.

From A to B the mixture is **compressed adiabatically.**

From B to C the mixture is **ignited** and burned at **constant volume.**

From C to D the gases **expand adiabatically.**

From D to A heat is **discharged** at **constant volume.**

From A to E the inert gases are swept out of the cylinder at **constant pressure** by the piston.

This description fits the four-stroke cycle. In the two-stroke cycle the operations EA and AE are performed in a separate cylinder, but as they do not enter into the thermal analysis, such an analysis for the four-stroke would apply also to the two-stroke cycle.

This cycle appears on the TS-plane as shown in Fig. 20. BC and AD are **constant-volume** lines and AB and CD are **adiabatics.** Heat is added from B to C and if we assume c_v to be constant, we have

$$Q_{bc} = Mc_v (T_c - T_b) \tag{1}$$

and the **heat rejected** from D to A

$$Q_{da} = Mc_v (T_d - T_a). \tag{2}$$

Heat transformed into **work** is the difference of these two or

$$Q_{bc} - Q_{da} = Mc_v\,(T_c - T_b - T_d + T_a), \tag{3}$$

$$E = \frac{Mc_v\,(T_c - T_b - T_d + T_a)}{Mc_v\,(T_c - T_b)} = 1 - \frac{T_d - T_a}{T_c - T_b}, \tag{4}$$

or
$$E = 1 - \frac{T_a}{T_b}\,\frac{\dfrac{T_d}{T_a} - 1}{\dfrac{T_c}{T_b} - 1}. \tag{5}$$

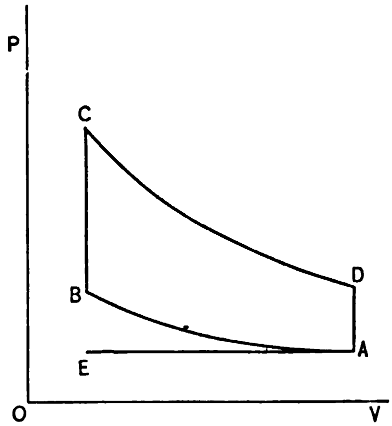


FIG. 19.

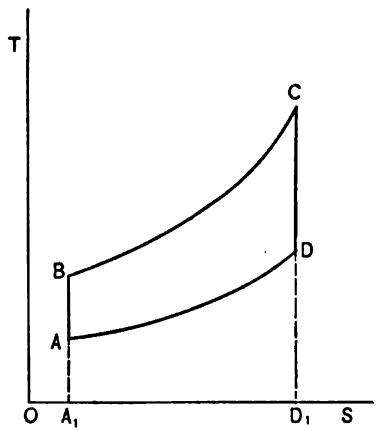


FIG. 20.

Otto **Diagram**.

From **adiabatic** relations we get

$$\frac{T_a}{T_b} = \left(\frac{V_b}{V_a}\right)^{n-1},$$

$$T_c = \left(\frac{V_d}{V_c}\right)^{n-1} T_d,$$

$$T_b = \left(\frac{V_a}{V_b}\right)^{n-1} T_a.$$

But as $V_b = V_c$ and $V_a = V_d$ $\therefore$ $\dfrac{V_a}{V_b} = \dfrac{V_d}{V_c}$ and $\dfrac{T_c}{T_b} = \dfrac{T_d}{T_a}$.

Then
$$E = 1 - \left(\frac{V_b}{V_a}\right)^{n-1} = 1 - \frac{1}{r^{n-1}}, \tag{6}$$

where $r = \dfrac{V_a}{V_b}$, the ratio of compression.

From equation (1) we get

$$\frac{T_c}{T_b} = \frac{Q_{bc}}{c_v T_b} + 1,$$

or

$$\frac{T_c}{T_a} = r^{n-1}\left(\frac{Q_{bc}}{c_v T_b} + 1\right), \tag{7}$$

which gives us the range of temperatures in the cycle.

Comparison of Cycles. — Of the five cycles analyzed, two are not used in practice. The **Carnot** is a **theoretical** cycle used only as a standard. To use this cycle and obtain its theoretical efficiency would require higher pressures and temperatures than could be handled in actual practice.

The efficiency of the **Lenoir** cycle, due to absence of compression, is so low that this cycle was discarded after a few years of use and has never been taken up again.

The **Brayton, Diesel** and **Otto** cycles are shown, superimposed, in Fig. 21. The **minimum temperature** at A and the **maximum temperature** at C are the **same** for all three. It is seen, then, that the **Brayton** cycle $AB'CD'$ has the largest area and the **Otto** cycle $ABCD$ the smallest. Hence, between the same temperature limits and the same maximum pressure P_c the **Brayton** cycle is the most efficient, and the **Otto** the least efficient. Comparing the maximum volumes, it is seen that the **Otto** and **Diesel** cycles have the same maximum volumes V_d while the **Brayton** cycle requires a greater volume, as shown by the point D'. The **Diesel** cycle, therefore, combines the advantages of the high efficiency of the **Brayton** cycle due to high compression pressure and the smaller cylinder volume of the **Otto** cycle.

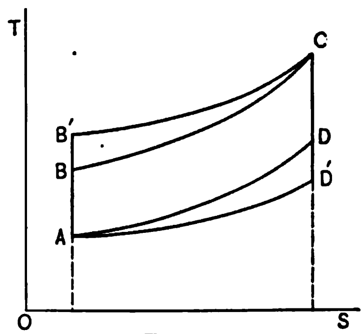


FIG. 21.

Further practical comparisons of the **Otto** and **Diesel** cycles will be discussed when the design of these two types is taken up later.

CHAPTER V

FUELS AND COMBUSTION

Combustion — Air Required — Gases — Hydrocarbons — Calorimeters — Specific Heat — Vapor Pressure

General. — **Carbon** and **hydrogen** are the elements that form combinations to furnish heat in gas-engine fuels. Some fuels contain, in combination with hydrogen and carbon, small percentages of oxygen. These fuels are usually artificial, such as producer gas and coal gas. When oxygen is present in coal, it is looked upon as a detriment, as it is already combined with hydrogen, and reduces the total heating value of the hydrogen content.

The principal **natural fuels** that are used for internal combustion engines are **coal, wood, peat, lignite, oil** and **natural gas.** Among the **artificial fuels** are **illuminating gas, blast-furnace gas, producer gas, coke oven gas, alcohol,** etc. **Producer gas** plays a very important part in gas-engine work. Indications seem to point to producer gas as the solution to the gas-engine problem, since, by the producer alone, can the low-grade fuels be made available for furnishing power efficiently.

Combustion. — The process of **combustion** is nothing more than the union of a fuel with oxygen to form an oxide. Slow and gradual union takes place, but is not brought under the term **combustion** as no sensible heat is developed. Thus, the rusting of iron is a union with oxygen, as is the decaying of animal and vegetable matter, but these chemical reactions do not come under the term combustion in the sense in which we wish to use it in connection with this work.

In a **furnace** or **stove** air is brought into contact with coal (carbon) at a high temperature. The oxygen of the air unites with the hot carbon and, if sufficient air be supplied, carbon dioxide results. If the air or oxygen supply be limited, the resulting product of combustion is mainly carbon monoxide. If the carbon burns to carbon dioxide, a certain amount of heat is generated, or liberated, for each pound of carbon consumed. If the air supply

is scant and the carbon burns to the monoxide, then a much smaller amount of heat will be generated. This, in general, illustrates perfect and imperfect combustion.

Combustion of Carbon. — The questions of finding heating values of gaseous mixtures and air required per unit, by weight or volume, may best be solved by use of the chemical or combustion formulæ. Thus, when carbon burns completely, we have

$$\begin{array}{ccc} C + O_2 & = & CO_2 \\ 12 \quad 32 & & 44 \end{array}$$

The numbers refer to the atomic weights of the elements, and they show that 12 parts of carbon, **by weight,** unite with 32 parts of oxygen to form 44 parts of carbon dioxide (CO_2). Thus, for each pound of carbon burned to CO_2, there are required $\frac{32}{12} = 2.67$ pounds of oxygen. The heat evolved during the burning of 1 pound of carbon is 14,600 British thermal units (B.t.u.).

For incomplete combustion we have

$$\begin{array}{ccc} 2\,C + O_2 & = & 2\,CO \\ 24 \quad 32 & & 56 \end{array}$$

Thus 24 parts of carbon unite with 32 parts of oxygen forming 56 parts of carbon monoxide (CO), or 1 pound of carbon requires 1.33 pounds of oxygen to burn to 2.33 pounds of CO. The heat generated in the partial combustion of 1 pound of carbon to CO is 4380 B.t.u. The difference $14{,}600 - 4380 = 10{,}220$ is still contained in the 2.33 pounds of CO. If this CO is lost, the heat, of course, is lost. If the CO is burned, the heat is regained. Thus we would expect the heating value of the CO to be $\frac{10{,}220}{2.33} =$ 4380 B.t.u. per pound, which is the figure usually given.

The equation for the combustion of the CO is

$$\begin{array}{ccc} 2\,CO + O_2 & = & 2\,CO_2 \\ 56 \quad 32 & & 88 \end{array}$$

which indicates that 1 pound of CO requires 0.572 pound of oxygen for combustion, resulting in 1.572 pounds of CO_2.

In the pound of CO there will be $\frac{12}{28}$ pound of carbon. This carbon requires 0.572 pound of oxygen to complete the combustion or $\frac{0.572}{12} \times 28 = 1.33$ pounds of oxygen per pound of carbon in the CO. Thus it requires the same amount of oxygen to burn

the carbon to CO and then to CO_2 as to burn the carbon directly to CO_2 in one operation.

Combustion of Hydrogen. — When hydrogen is burned we have the chemical reaction

$$\underset{4}{2\,H_2} + \underset{32}{O_2} = \underset{36}{2\,H_2O}$$

showing that 1 pound of hydrogen unites with 8 pounds of oxygen, forming 9 pounds of water.

When this process is reversed and steam is forced through a bed of incandescent carbon, the opposite reaction results. The steam is broken up into hydrogen and oxygen and the latter unites with the carbon in the coke forming carbon monoxide.

For that reaction the equation is

$$\underset{18}{H_2O} + \underset{12}{C} = \underset{28}{CO} + \underset{2}{H_2}$$

showing that 1 pound of water unites with 0.667 pound of carbon, resulting in 1.556 pounds of carbon monoxide and 0.111 pound of hydrogen.

When hydrogen is burned to water at 32° F., the heat generated per pound of hydrogen is 62,100 B.t.u. When the reverse process is used, that is, when water is dissociated, for every pound of hydrogen that results 62,100 B.t.u. must be supplied. Since it requires 9 pounds of water to be dissociated to form 1 pound of hydrogen, the heat necessary to dissociate 1 pound of water is 6900 B.t.u.

Combustion of Hydrocarbons. — Some of the gas-engine fuels, natural and illuminating gas and the producer gas made from bituminous coal, contain **heavy hydrocarbons.** The principal ones of these are

Methane or **marsh gas** CH_4.

Ethylene or **olefiant gas** C_2H_4.

Both these gases burn to carbon dioxide and water and their chemical equations may be written as for carbon and hydrogen alone. For methane we have

$$\underset{16}{CH_4} + \underset{64}{2\,O_2} = \underset{44}{CO_2} + \underset{36}{2\,H_2O}$$

It is evident that 1 pound of methane requires 4 pounds of oxygen for combustion, resulting in $2\frac{3}{4}$ pounds of carbon dioxide

and $2\frac{1}{4}$ pounds of water. The heat generated in the combustion of 1 pound of the gas is 24,000 B.t.u.

When ethylene is burned the equation is

$$\underset{28}{C_2H_4} + \underset{96}{3\ O_2} = \underset{88}{2\ CO_2} + \underset{36}{2\ H_2O}$$

indicating that 1 pound of C_2H_4 requires 3.42 pounds of oxygen, forming 3.14 pounds of CO_2 and 1.28 pounds of water. The heat generated is about 21,500 B.t.u. per pound.

A **general equation** for the oxygen required for the combustion of any hydrocarbon may be written:

$$2.667\,C + 8\,H,$$

where C and H are the proportions of carbon and hydrogen respectively entering into the composition.

Atmospheric Air. — In all forms of internal combustion engines the oxygen required for the combustion of the fuel is obtained from the atmosphere. Since air contains only a small proportion of oxygen, we must carry our calculations one step further and find the quantity of air required for combustion of the various fuels.

Air is composed principally of oxygen and nitrogen in the following proportions:

By weight		By volume	
Oxygen...........	0.231	Oxygen............	0.21
Nitrogen.........	0.769	Nitrogen..........	0.79
	1.000		1.00

The other constituents of the atmosphere, CO_2, water vapor, etc., do not form a large enough proportion to have any influence on gas-engine calculations.

When the weight of oxygen required for combustion has been found, only a short calculation is required to find the weight of air corresponding. Thus, for the complete combustion of one pound of carbon, 2.67 pounds of oxygen are required. The air necessary to furnish this oxygen, since air contains 23 per cent oxygen by weight, will be $\frac{2.67}{0.23} = 11.6$ pounds.

For the combustion of hydrogen, 8 pounds of oxygen are required, necessitating the use of $8 \div 0.23 = 34.63$ pounds of air.

Then for the complete combustion of methane which is 75 per cent carbon and 25 per cent hydrogen, there would be required

$$0.75 \times 11.6 + 0.25 \times 34.63 = 17.36 \text{ pounds of air.}$$

When fuels, such as alcohol, contain oxygen, this oxygen may be subtracted from the theoretical oxygen required, as the oxygen in the fuel unites with some of the hydrogen.

Volume of Air Required for Combustion. — The volume of 1 pound of air at 32° F. is 12.39 cubic feet and at 62° F. is 13.14 cubic feet. After the weight of air has been determined, the volume at 32 degrees or 62 degrees may be found by multiplying by 12.39 or 13.14. Thus, the 11.6 pounds of air required to burn a pound of carbon to CO_2 would occupy a space of 143.7 cubic feet at 32 degrees, atmospheric pressure, or 152.4 cubic feet at 62 degrees. It would be possible at this point to derive formulæ showing the volume of air at any temperature required for any fuel containing C, H_2 and O_2. These formulæ, however, are easily forgotten if not used constantly, so that reference must be made to some book containing the formulæ every time they are to be used. The author prefers to use the combustion equations each time that the air required for a fuel is to be calculated. In that way bulky formulæ are done away with, and the chance for errors to creep in, due to the misuse of these formulæ, is very slight.

Air Required for a Compound Gas. — The air required for the combustion of a pound of compound gas may be found in the same way that it was found for methane and ethylene. If no tables are at hand, the combustion equation may be written for each constituent and the oxygen required **per pound** of this constituent found. If this value is then multiplied by the percentage by weight of that constituent in the mixture, and these products added, the total will be the **oxygen** required per pound of the compound. This can easily be changed into pounds or cubic feet of **air**.

Example: Find the pounds of air required per pound of blast-furnace gas of the following composition:

	Per cent
CO	27
H_2	3
CO_2	12
N_2	58
	100

In order to use the chemical equations, this analysis must be reduced to the weight basis. If each percentage, divided by 100, is multiplied by the molecular density of the corresponding gas, the results will give the weight of each constituent compared to the weight of a molecule of hydrogen. If these individual weights be divided by their total, the quotient will be the percentage weight of that constituent.

$$
\begin{array}{rlr}
CO = 0.27 \times 14 = & 3.78 & 25.90 \\
H_2 = 0.03 \times \ \ 1 = & 0.03 & 00.21 \\
CO_2 = 0.12 \times 22 = & 2.64 & 18.10 \\
N_2 = 0.58 \times 14 = & \underline{8.12} & \underline{55.79} \\
 & 14.57 & 100.00
\end{array}
$$

Since the CO and the hydrogen are the only combustibles in this gas, we have only to find the air required for them. We have already found that 1 pound of CO requires 0.572 pound of oxygen for combustion and that 1 pound of hydrogen requires 8 pounds of oxygen. The total oxygen per pound of this gas will be 0.165 pound.

$$
\begin{array}{lr}
0.572 \times 0.259 = & 0.148 \\
8 \quad\ \ \times 0.0021 = & \underline{0.017} \\
 & 0.165
\end{array}
$$

The air to correspond will be $\frac{0.165}{0.23} = 0.72$ pound.

Air Per Cubic Foot of Gas. — In gas-engine design it is frequently necessary to find the volume of air required per cubic foot of gas. If the density of the gas is known, the relative volumes of air and gas may be found from the relative weights found in the preceding article. A more convenient method is to use a table giving the cubic feet of air required per cubic foot of gas for the different fuel gases. Such tables are given here, Tables I and II. Using Table II we find that 1 cubic foot of CO requires 2.4 cubic feet of air for combustion and a cubic foot of hydrogen the same. Using the volumetric analysis, we find the air required per cubic foot of the gas mentioned above is 0.72 cubic foot.

$$
\begin{array}{lr}
CO\ 0.27 \times 2.4 = & 0.648 \\
H_2\ 0.03 \times 2.4 = & \underline{0.072} \\
 & 0.720
\end{array}
$$

Volume of Products of Combustion. — It is sometimes of importance to find the **volume** of **products of combustion** of a gas.

The weight of the products of combustion of the fuel is 1 pound, plus the weight of air supplied per pound of fuel.

TABLE I

DATA ON COMMON FUELS, ETC.

	Chemical symbol.	Molecular weight.	Weight per cu. ft. at 14.7 lbs. pressure and		Calorific value.					
					B.t.u. per lb.		B.t.u. per cu. ft. of gas.			
							High value at		Low value at	
			32° F.	62° F.	High.	Low.	32° F.	62° F.	32° F.	62° F.
Hydrogen	H_2	2	0.005591	0.005269	62,100	52,275	347	327	292	275
Methane (marsh gas)	CH_4	16	0.04473	0.04215	24,000	21,544	1073	1013	968	910
Ethylene (olefiant gas)	C_2H_4	28	0.07827	0.07376	21,900	20,496	1714	1615	1600	1512
Acetylene	C_2H_2	26	0.07268	0.06849	21,860	21,094	1592	1500	1540	1440
Ethane	C_2H_6	30	0.08386	0.07903	22,340	20,375	1880	1800	1700	1620
Butylene	C_4H_8	56	0.15655	0.1476	20,860	19,405	3264	3080	3036	2860
Benzene	C_6H_6	78	0.21805	0.2055	18,100	17,343	3940	3720	3780	3560
Carbon monoxide	CO	28	0.07827	0.07376	4,380	4,380	344	324	344	324
Carbon	C	12	......	......	14,600	14,600	....	....	....	....
Carbon dioxide	CO_2	44	0.123	0.1159	......	......	....	....	....	....
Nitrogen	N_2	28	0.07827	0.07376	......	......	....	....	....	....
Oxygen	O_2	32	0.08936	0.08421	......	......	....	....	....	....

TABLE II

DATA ON COMMON FUELS

	Required weight of air per lb. of fuel, lbs.	Weight of combustion products per lb. of gas.			Volume of combustion products per lb. of gas at 32° F.			Cu. ft. of air required for combustion.	
		CO_2, lbs.	H_2O, lbs.	N_2 lbs.	CO_2 cu. ft.	H_2O cu. ft.	N_2 cu. ft.	Per lb. of gas at 62° F.	Per cu. ft. gas.
Hydrogen	34.64	....	9	26.64	......	178.87	340.46	456	2.4
Methane (marsh gas)	17.32	2.75	2.25	13.32	22.36	44.72	170.23	228	9.61
Ethylene (olefiant gas)	14.85	3.14	1.29	11.42	25.55	25.58	145.91	195	14.3
Acetylene	13.32	3.39	0.69	10.24	27.52	13.75	130.94	175	12.0
Ethane	16.16	2.93	1.80	12.43	23.85	35.77	158.87	213	16.83
Butylene	14.85	3.14	1.29	11.42	25.55	25.58	145.91	195	28.8
Benzene	13.32	3.39	0.69	10 24	27.52	13.75	130.94	175	36.0
Carbon monoxide	2.48	1.57	....	1.91	12.78	......	24.35	33	2.4
Carbon	11.56	3.67	....	8.89	29.81	......	113.59	152	......

To determine the total volume of the combustion products some standard of temperature and pressure must be decided on. If we select 32° F. and atmospheric pressure, then a pound of CO_2 will occupy 8.13 and a pound of nitrogen 12.78 cubic feet. The H_2O

vapor formed by the combustion of hydrogen would condense if the temperature were lowered to 32° F. at atmospheric pressure. If we consider this vapor to be a gas at that temperature, then the volume of a pound would be 19.27 cubic feet.

Using these values we will proceed to find the volume of the products of combustion of 1 pound of methane CH_4. From the preceding articles we found that a pound of carbon requires 2.67 pounds of oxygen for combustion and a pound of hydrogen 8 pounds of oxygen. Also, from the molecular weight it is evident that methane is composed of 75 per cent carbon and 25 per cent hydrogen by weight. The volume of the combustion products will then be

$$0.75\ (1 + 2.67)\ \ 8.13 = 22.38 \text{ for the } CO_2,$$
$$0.25\ (1 + 8.0\ )\ 19.27 = 43.36 \text{ for the } H_2O,$$

$(0.75 \times 2.67 + 0.25 \times 8.0\ )\ 12.78 \times 3.33^* = 170.34$ cubic feet at 62° F. for the nitrogen.

It is often necessary to find the volume of the exhaust gases at a higher temperature than the standard, 32 degrees. To do this we may use the law that, at constant pressure, the volume of the gas varies as the absolute temperature. Then the volume at any temperature "t" would be:

$$\text{Vol. at ``}t\text{'' °F.} = \frac{460 + t}{460 + 32} \times \text{vol. at 32° F.}$$

At any temperature below 212 degrees, when the pressure is atmospheric, the volume of the water vapor would contract to 0.0016 of the volume computed for 32 degrees, so that for any temperature below the boiling point we may assume the volume of the water vapor to be zero.

Heating Value of Fuels Containing Hydrogen. — We have seen that 1 pound of hydrogen unites with 8 pounds of oxygen, forming 9 pounds of water, and generates 62,100 heat units in forming the combination. This heating value is found in a calorimeter by burning the gas at constant pressure. The heat is absorbed by the products of combustion as sensible heat and by the water vapor as latent heat. To find the heating value at a standard of 32 degrees, the products of combustion are cooled to that point, and as that point is far below the boiling point of water at

* This factor shows the relation of nitrogen to oxygen in the air; there are 3.33 pounds of nitrogen present for every pound of oxygen.

atmospheric pressure, the latent heat of the vapor is given up and is measured as a part of the 62,100 B.t.u.

If, in using the calorimeter, we did not cool the products of combustion below 212 degrees, the presence of the latent heat would not become manifest and the result would be a lower heating value determined per pound of hydrogen than if the products of combustion were cooled below 212 degrees. Similarly, when the heat of combustion is used for heating the charge in a gas-engine cylinder, the total heat of the hydrogen is generated, but if the exhaust leaves the engine at a higher temperature than 212 degrees, as it always does, the latent heat of the water vapor is carried with it and has had no part in producing work in the cylinder. When the water vapor formed from the hydrogen in the fuel is not condensed the heat generated per unit of fuel is termed the lower heating value.

The question of which heating value to charge up to a gas-engine in finding the thermal efficiency has often been raised. It seems fair to charge the engine with the higher heating value, even though the engine does not use all of it. The latent heat of the steam is charged against the steam-engine, so should the latent heat of the water vapor be charged against the gas-engine.

In some instances, in studying the effects of heat on the charge in the engine cylinder, it will be necessary to use the lower heating value. This question will arise in determining the size of the cylinder to generate a given power.

Any gas fuel containing hydrogen will have a **higher** and a **lower** heating value, and the latter is always found by subtracting from the former the latent heat in the steam formed by the combustion of hydrogen in the fuel. If we take the standard at 32 degrees, we must subtract from the higher heating value the latent heat of the steam at 32 degrees. This may be obtained from Regnault's formula,

$$H = 1091.7 - 0.695\,(t - 32);$$

or at 32 degrees this becomes 1091.7. As 1 pound of hydrogen forms 9 pounds of water at combustion, we must subtract $9 \times 1091.7 = 9825$ from 62,100, giving 52,275 as the lower heating value.

If, in any hydrocarbon, the ratio of hydrogen expressed in decimals be x, the vapor formed at combustion of the gas will be $x \times 9$

pounds per pound of gas, and the heat of vaporization will be $x \times 9 \times 1091.7$. Therefore, $x \times 9825$ heat units should be subtracted from the higher calorific value of the gas in order to get the lower.

The Gas Calorimeter. — Calorimeters, in general, are arranged for burning in the apparatus a known quantity of fuel with oxygen or air for combustion, and absorbing the heat by means of a measured quantity of water. If the increase in temperature of water is observed, the heat generated per pound of fuel will be obtained from the equation

$$Q = \frac{W(T_2 - T_1)}{L},$$

where W is the weight of the cooling water, $T_2 - T_1$ its increase in temperature and L the weight of the fuel burned.

The Junker's calorimeter is one of the best known for finding the heating value of gases. This instrument is shown in Fig. 22. The gas coming from the meter is burned in a Bunsen burner placed in the cylindrical combustion chamber of the instrument. The hot gases rise in a central chamber, pass down through a series of small tubes in the annular water jacket surrounding the combustion chamber, unite again in a single flue at the bottom of the instrument and pass out through an orifice whose size may be regulated by a damper. Water enters through a rubber tube the small elevated tank i which serves to supply water to the calorimeter under a constant head. The amount of water flowing into the instrument is controlled by a valve a and the waste water from i is discharged at c. The water traversing the instrument passes out at d and during the actual test is measured in a graduated cylinder, or collected in a vessel, and subsequently weighed.

The incoming water passes the thermometer which registers the inlet temperature. The water then descends in a small pipe within the outer casing to the bottom of the instrument. The water next rises in the annular chamber surrounding the gas passages, thus passing in a direction opposite to that of the combustion gases. The water from the annular chamber passes through the drum provided with baffle plates to mix it and insure that all parts of the stream are of uniform temperature, passes the thermometer which registers the outlet temperature, and out.

The water formed in the combustion of the hydrogen and the hydrocarbons in the gas is condensed as it descends the annular

condenser and passes out the drip and is caught in a graduated cylinder. The working parts of the calorimeter thus described are insulated from the room by an air jacket. The outer casing of the calorimeter is nickel-plated and is polished to lessen the loss of heat by radiation.

The Mahler Bomb Calorimeter.—For solid or liquid fuels the **Mahler bomb calorimeter** is frequently used. It consists of a

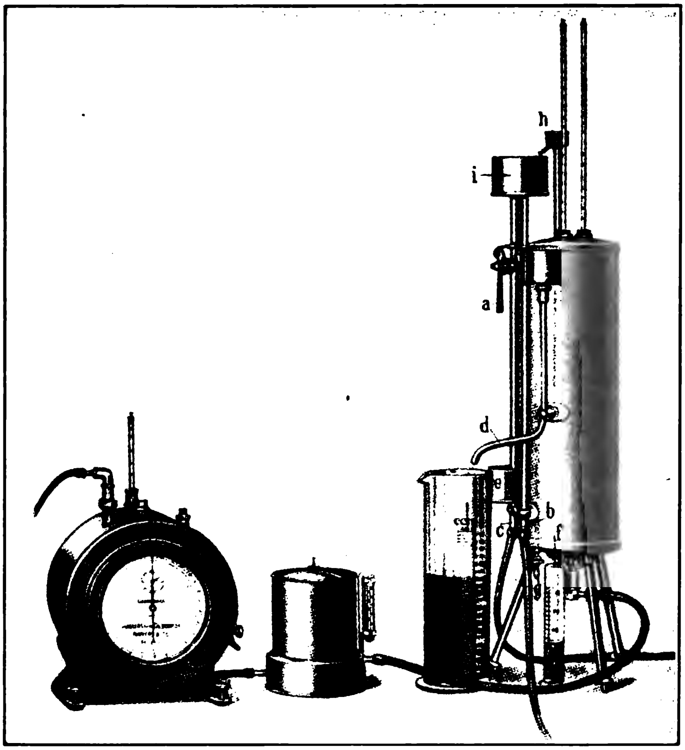


FIG. 22.—The Junker Gas Calorimeter.

small steel vessel or bomb lined, usually, with an enamel on which acids will have no effect. In this bomb a small known weight of fuel is burned in an atmosphere of dense oxygen, while the instrument is submerged in a known quantity of water which absorbs the heat of combustion.

On the cap of the bomb is suspended a platinum pan on which is placed the fuel to be tested. If it is coal, it must be finely

powdered to insure complete combustion. After the coal and platinum pan are placed in position, a fine iron fuse wire is attached to two terminals in the cap, and passed through the sample of coal. After the coal and fuse wire are in place, suspended from the cap, this cap is screwed tightly on the bomb, leakage being prevented by a lead gasket. The bomb is then charged with about 20 atmospheres of oxygen, placed in the can of carefully weighed water and the firing wires are attached to the terminals. The can containing the water is insulated so that little or no heat will be transmitted to the outer air. In order to measure the heat thus transmitted, the can and insulation are placed in a second receptacle containing water. By measuring the rise in temperature of this water, the radiation may be found.

Since the sample of coal used is small, about 0.6 gram, and the water in which the bomb is submerged is usually about 2000 grams, the rise in temperature will be very small, from 1.5° to 3° C. Very sensitive thermometers must be used to measure the rise, and readings must be carefully taken.

While this is just an outline of the procedure when a bomb calorimeter is used, a more thorough explanation would be burdensome unless the reader is especially interested. If tests are to be made with such a calorimeter, books on fuel analysis should be consulted for methods in computing radiation, corrections for specific heat of water, sulphur in coal, heating of fuse wire, etc.

Heating Value of Fuel by Formula. — In many cases the chemical analysis of a fuel is obtained more easily than a calorimeter test, and the heating value of a solid fuel may be approximated very closely by use of this analysis.

One well-known formula used for this work is credited to Dulong:

$$Q = 14{,}600\,\mathrm{C} + 62{,}100\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right), \tag{1}$$

where Q is the heating value of the fuel, B.t.u. per pound, C the percentage carbon, H the percentage hydrogen and O the percentage oxygen. C, H and O are to be expressed in decimal parts of the total weight of the fuel as a unit.

This formula is based on two important laws by Dulong:

"The heating value of a compound fuel is the sum of the heating values that would be obtained by the combustion of each component separately,

providing that the fuel does not contain oxygen and hydrogen chemically combined with it."

"When oxygen and hydrogen do exist in chemical combination with a fuel, it is only the part of the hydrogen outside of that required for forming water with the oxygen that will evolve heat; and the oxygen required for combustion will be reduced from the amount the analysis would show by the amount of oxygen in the fuel."

For liquid hydrocarbons the following formula seems to give results which check the calorimeter tests closely, and consequently it is often used:

$$Q = 14{,}600\,\mathrm{C} + 52{,}230\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right). \tag{2}$$

If formula (1) be applied to methane, for instance, the result will be

$$\begin{aligned} \mathrm{C} &= 0.75 \times 14{,}600 = 10{,}950 \\ \mathrm{H_2} &= 0.25 \times 62{,}100 = \underline{15{,}500} \\ &\qquad\quad \text{Total} \quad 26{,}450 \end{aligned}$$

and by formula (2) we get

$$\begin{aligned} \mathrm{C} &= 0.75 \times 14{,}600 = 10{,}950 \\ \mathrm{H_2} &= 0.25 \times 52{,}230 = \underline{13{,}058} \\ &\qquad\quad \text{Total} \quad 24{,}008 \end{aligned}$$

The first formula gives a result that is much higher and the second a result that checks the heating value given for methane in Table I. This difference is due to the fact that the heat to break up the chemical union of the carbon and hydrogen is not available, that the actual heat evolved is the theoretical heating value less the heat of the chemical union; hence the discrepancy in the results found above.

Specific Heat and Flame Temperature. — The specific heat of a substance has already been defined as the quantity of heat necessary to raise the temperature of a unit weight of that substance one degree, the specific heat of water being unity.

Gases have two specific heats. If a gas is heated at constant volume, no external work is done and consequently no heat need be supplied to do this work. If heat is added while pressure is constant, then the volume increases, work is done and heat must be added to do this work. Therefore the specific heat at constant pressure is greater than the specific heat at constant volume.

The specific heats of some of the more important gases are given in Table III. These are the values found by Regnault.

TABLE III

SPECIFIC HEATS OF GASES AT 212° FAHRENHEIT

	At constant pressure, c_p.	At constant volume, c_v.	$\frac{c_p}{c_v}$.
Air	0.2375	0.1700	1.41
Superheated steam H_2O	0.4805–	0.3585	1.34
Hydrogen H_2	3.4090	2.4177	1.41
Oxygen O_2	0.2175	0.1543	1.41
Nitrogen N_2	0.2438	0.1729	1.41
Carbon monoxide CO	0.2479–	0.1738	1.41
Carbon dioxide CO_2	0.2169	0.1570	1.29
Marsh gas CH_4	0.5929	0.4505	1.316
Olefiant gas C_2H_4	0.4040	0.3200	1.26
Average anthracite gas	0.24 to 0.26	....	....
Average bituminous gas	0.24 to 0.26	....	....
Average blast-furnace gas	0.23	....	....
Average chimney gas	0.24	....	....

The maximum temperature of the products of combustion of a fuel, or, in other words, the **flame temperature,** may be approximated by assuming that all the heat of combustion is used to heat the products of combustion, and that the specific heat of the products is constant. Then the temperature rise resulting from the combustion would be found by the following formula:

$$T = \frac{Q}{CW},$$

where Q is the heating value of the fuel per pound, C is the mean specific heat of the products of combustion and W is the weight of the products of combustion per pound of the fuel. The actual temperature after combustion will be the sum of the original and the rise.

Example: Find the flame temperature of the blast-furnace gas whose composition by weight is CO = 22 per cent, H_2 = 1 per cent, CO_2 = 14 per cent and N_2 = 63 per cent. Excess air is 15 per cent. Combustion is to take place at atmospheric pressure.

The first step is to find the air required for perfect combustion. This is found by the help of Table II.

$$\begin{aligned} \text{Air for CO} &= 0.22 \times 2.477 = 0.5460 \text{ pound} \\ \text{Air for } H_2 &= 0.01 \times 34.64 = \underline{0.3464} \text{ pound} \\ &\qquad\qquad \text{Total} \quad 0.8924 \text{ pound} \\ \text{Excess} &= 0.15 \times 0.8924 = 0.1340 \text{ pound.} \end{aligned}$$

Total weight of combustion products will be

$$0.8924 + 0.1340 + 1.000 = 2.0264.$$

The weight of the products of combustion, per pound of gas, may also be found from the same table.

	Weights of products of combustion.			
	H_2O	CO_2	N_2	Air.
From CO, 0.22 pound	....	0.348	0.419	....
From hydrogen, 0.01 pound	0.09	....	0.267	....
From CO_2	....	0.140	....	....
From nitrogen	....	....	0.630	....
From excess air	....	....	....	0.134
Totals	0.09	0.488	1.316	0.134

To find the specific heat of the mixture, each component must be multiplied by its specific heat, the products added and divided by the total weight. For this gas we have

H_2O	0.09×0.481	$= 0.0433$
CO_2	0.488×0.217	$= 0.1059$
N_2	1.316×0.244	$= 0.3135$
Air	0.134×0.238	$= 0.0319$
	2.028	0.4946

This value 2.028 should check the value 2.0264 found above.

The mean specific heat of the combustion products will be:

$$\frac{0.4946}{2.028} = 0.245.$$

If the heating value of the fuel is 1350 B.t.u. per pound, then the theoretical temperature of the flame will be

$$\frac{1350}{2.028 \times 0.245} = 2730^\circ \text{ F. above the atmospheric temperature.}$$

If the gas is burned at constant volume, the specific heat at constant volume would have to be used to find the flame temperature. This is found to be .175 for the gas chosen by the same method used above forgetting the value at constant pressure, and as it is lower than the specific heat at constant pressure, the corresponding temperature will be higher, or

$$T_v = \frac{1350}{0.175 \times 2.018} = 3800 \text{ degrees}$$

above atmospheric temperature. Since, in the engine cylinder, not all the heat of combustion is utilized in raising the temperature of the products of combustion, this temperature is never approached in practice.

Density of Gases. — Avogadro's law states the fact that equal volumes of all gases at the same temperature and pressure contain the same number of molecules, and as a consequence it follows that the ratio between the weights of equal volumes, or between the weights of unit volumes, of any two gases is as the ratio of their molecular weights.

The molecular weights of substances are usually referred to the weight of the hydrogen molecule as a unit. Thus, since the molecular weight of hydrogen is 2, if we call that of any other gas m, the ratio of the weight of unit volumes of hydrogen and the given gas will be as 2 to m or as 1 to $\frac{m}{2}$. Hence, the weight of a unit volume of a gas, or the density, is

$$\text{density of a gas} = \frac{m}{2} \times \text{density of hydrogen.}$$

To determine the weight per cubic foot of any gas it will only be necessary to multiply the density of hydrogen by half the molecular weight of the gas in question. The weight per cubic foot of hydrogen is, at 32° F. and 14.7 pounds (atmospheric) pressure, 0.005591 pound. At the same standard conditions the number of cubic feet in a pound of hydrogen will be the reciprocal of 0.005591 or 178.87.

As an example, the weight per cubic foot of marsh gas CH_4 will be $\frac{16}{2} \times 0.005591 = 0.044728$.

The density of a gas at constant pressure varies inversely as the absolute temperature. Thus at t degrees the density will be

$$D_t = \frac{459.2 + 32}{459.2 + t^{\circ}} D_{32}.$$

It may sometimes be more convenient to refer the density of a gas to the density of air instead of hydrogen. As dry air at a temperature of 32° F. and a pressure of 14.7 pounds is $\frac{0.08728}{0.005591}$ $= 14.44$ times the weight of the hydrogen, then density of a gas referred to air $= \frac{m}{28.88}$ where m is the molecular weight of the gas.

The volume per pound of gas is the reciprocal of the density and since we have

$$\text{density of a gas} = \frac{m}{2} \times 0.005591,$$

then

$$\frac{1}{\text{density of gas}} = \frac{2}{0.005591\ m} = \frac{357.74}{m}.$$

That is, to find the cubic feet per pound of a given gas, divide 357.74 by the molecular weight of that gas.

Heating Value Per Unit Volume. — It is usually more convenient to measure the fuel supply of a gas-engine in cubic feet than in pounds. Then, instead of using the heating value per pound of the gas, it will be necessary to use the heating value per cubic foot. If the heating value per pound be known, the heating value per cubic foot may be found by multiplying the B.t.u. per pound by the density of the gas.

To reduce the volume of gas to a standard condition, usually 32 degrees and atmospheric pressure, the following formulæ will be convenient. The subscript x will be used to designate any condition and s the standard condition.

T_x = any absolute temperature = $t_x + 459.2°$ F.,
P_x = any absolute pressure = 14.7 + gauge pressure,
V_x = corresponding volume, in cubic feet,
T_s = standard absolute temperature = 491.2° F.,
P_s = standard pressure = 14.7 pounds per square inch,
V_s = volume of gas at standard temperature and pressure,

then

$$V_s = V_x \frac{P_x}{P_s} \frac{T_s}{T_x} = V_x \frac{491.2\,P_x}{14.7\,T_x}.$$

It is apparent that the denser the gas, the greater its heating value per cubic foot, or the heating value varies directly with the pressure and inversely as the temperature. Thus, if the heating value of hydrogen be 347 B.t.u. per cubic foot at 32° F. and atmospheric pressure, its heating value at 62 degrees and the same pressure will be $347 \times \frac{491.2}{521.2} = 327$ B.t.u.

At 212° F. and 14 pounds pressure, absolute, the heating value would be $347 \times \frac{491.2}{671.2} \times \frac{14.0}{14.7} = 241.5$ B.t.u.

Vapor Pressure. — If the laws of perfect gases hold with fuel vapors, then the weight of fuel vapor present in a given volume, at a fixed temperature, is proportional to its vapor pressure. There is a definite limit to the amount of condensable vapor that can

exist in a given space, at any given temperature. There is, for each temperature of the vapor, a limit to the pressure that it can sustain without becoming partially condensed back into the liquid.

In raising the temperature of a vapor its maximum vapor pressure, or saturation pressure, is increased and for each pressure there is a certain temperature where partial condensation begins. This temperature is called the saturation temperature at that given pressure. If the temperature of a saturated vapor is lowered, then the vapor becomes moist, and if its temperature is raised, it becomes superheated.

In Table IV is given the vapor pressures of saturation for different motor fuels and water. The values for the alcohols are from the "Fifth Revised Edition of the Smithsonian Physical Tables" and those for benzol, hexane and gasoline from Sorel's "Carburation et Combustion dans les Moteur a Alcool."

When two gases are enclosed in the same receptacle and subjected to an external pressure, both help to sustain that pressure in the ratio of their respective vapor pressures, and the sum of the vapor pressures of the two gases must equal the pressure to which they are subjected.

TABLE IV

VAPOR PRESSURE OF SATURATION

Temperature.		Vapor pressure of saturation in millimeters of mercury.						
							Gasoline.	
Degs. C.	Degs. F.	Pure ethyl alcohol C_2H_6O.	Pure methyl alcohol CH_4O.	Benzol.	Water.	Hexane C_6H_{14}.	Automobiline. Sp. density at 62° F., 0.703.	Stelline. Sp. density at 62° F., 0.673.
0	32	12.7	27	27	5	45	99	164
5	41	17.6	37	36	7	58	115	190
10	50	24.2	50	45	9	74	133	220
15	59	33.0	67	61	13	95	154	255
20	68	44.5	89	77	17	119	179	296
25	77	59.4	116	96	24	154	210	358
30	86	78.5	150	120	32	184	251	433
35	95	102.9	192	156	42	228	301	512
40	104	133.7	244	188	55	276	360	596
45	113	172.2	306	224	71	335	422	685
50	122	219.9	382	271	92	401	493	792
55	131	278.6	472	326	117	482	561	....
60	140	350.2	580	390	149	567	648	....
65	149	436.9	707	468	187	674	739	....

In a saturated mixture of fuel vapor and air, if we assume it to follow the laws of perfect gases, the ratio of volumes of vapor and air present in a given volume equals the direct ratio of their vapor pressures.

When the vapor pressure of a fuel is, under ordinary atmospheric conditions, much weaker than that of air, it may happen that the proportion of fuel to air is too small to form an explosive mixture, or the fuel mixture is too weak to be explosive. This is true of alcohol and kerosene and shows why these fuels must be heated before their vapor pressure is high enough to form an explosive mixture. With lighter fuels, such as benzine and gasoline, this is not the case. Their vapor pressures are so high that a saturated mixture under atmospheric conditions may be too rich to explode and must be diluted. This point will be taken up later when the common fuels are discussed further.

EXERCISES

1. A fuel contains by weight:

	Per cent
H_2	26
CO	40
N_2	Balance

Find higher heating value per pound and weight of air required for complete combustion.

2. Find lower heating of same.

3. Given a fuel containing by weight:

	Per cent
H_2	10
CO	45
CH_4	6
C_2H_4	4
CO_2	12
N_2	Balance

Find volume of air required per pound of gas.

4. Find the volume of combustion products per pound of the gas given in Prob. 3.

5. Given an analysis by volume as follows:

	Per cent
CO_2	12
H_2	2
CH_4	3
CO	27
N_2	56

Find analysis by weight.

6. (*a*) Use the Dulong formula to find the heating value of coal containing 64 per cent C, 5 per cent H_2 and 3.2 per cent O_2. (*b*) Use it for finding the heating value of methane and ethylene and check these values by Table I.

7. Find the flame temperature of gas in Prob. 5, gas to be burned at constant pressure. No excess air.

8. Find the heating value per cubic foot of gas in Prob. 5 when the gas temperature and pressure are 100° F. and 5 lbs. per sq. in. gauge, respectively.

CHAPTER VI

GAS-ENGINE FUELS IN LIQUID FORM — RELATION OF FUEL TO THE SIZE OF THE CYLINDER — PETROLEUM FUELS — GASOLINE — KEROSENE — ALCOHOL — DENATURED ALCOHOL

Characteristics of the Mixture After it is Drawn into the Cylinder. — The **maximum power** of a gas-engine depends on the mixture or charge after it is drawn into the cylinder. To increase the power of a **steam-engine,** the **cut-off** is increased until the engine is taking steam at full stroke; or the steam pressure may be increased. The matter is not so simple with the gas-engine. When the cylinder is filled with the proper explosive mixture, of maximum density, then the maximum power is available and we have no means of increasing this power. This accounts for the reluctance of the manufacturers to guarantee a large overload for their engines, and the absence of an overload when a brake test is made. If an engine does give a large overload, it shows that the rated load is smaller than it should be; that at the "rated load" the engine is really running at a lighter load than it should to give the maximum efficiency.

There are at least four conditions which influence the charge of fuel, or the mixture. These are: 1, the amount of air in the charge; 2, the temperature of the charge after it is drawn into the cylinder; 3, the pressure of the charge after it is drawn in; 4, the neutrals left in the cylinder after the exhaust stroke has been completed, which mix with the entering charge.

The amount of air in the charge should be carefully regulated by proper valves before the charge passes the inlet-valve of the engine. For gaseous fuels about 15 per cent above the theoretical air required for combustion should be used at full load. The temperature of the charge depends on the type of engine and on the load. The walls and valves of an air-cooled engine will be hotter than those of a water-cooled engine. If an engine is running at full load, the cylinder and valves will be hotter than if the engine is running at a lighter load. As the charge is drawn

in, at practically constant pressure, the hot walls and valves raise its temperature and decrease its density and its heating value proportionately. Another source of heat is the **inert gas** that is left in the cylinder. The quantity of heat given up by this inert gas depends on the amount of such gas left in the cylinder. This amount is the volume of the combustion space, or the clearance volume, and is greater on low-compression than on high-compression engines.

The **pressure reduction** of the charge, as it is drawn into the cylinder, depends on the amount of inlet-valve opening, or the amount that the governing valve is open, when the engine is governed by throttling. On large, slow-speed engines the drop in pressure should not be more than 1 pound at full load. On high-speed engines the drop exceeds this and this drop is often responsible for loss of power at high speeds, particularly in automobile engines.

In the following discussion the subscript "*a*" will be used to denote the condition of the charge after it has been drawn into the cylinder, and "*o*" the atmospheric conditions. If we assume that the charge is throttled 1 pound as it is drawn into the cylinder, its density will be made less in the ratio of its absolute pressures, or

$$d_a = \frac{p_a}{p_0} d_0 = \frac{13.7}{14.7} d_0 = 0.932\, d_0,$$

where d is the density and p the pressure.

In his book on "The Modern Gas Engine and Gas Producer" Mr. A. M. Levin has computed the probable temperatures of the charge after the completed suction stroke. He has assumed wall and valve temperatures and computed the amount of heat that would flow into the charge during the time the charge was in contact with the walls, valves, etc. He also computed the rise in temperature due to the neutrals that remain in the cylinder. The results of these calculations are as follows when the gas and air arrive at the inlet valve with a temperature of 62°:

	Water-cooled exhaust valve and piston.	Non-water-cooled exhaust valve and piston.
$r = 4,$	$t = 103°$ F.	$t = 120°$ F.
$r = 7,$	$t = 89°$ F.	$t = 107°$ F.

where

$$r = \frac{V_a}{V_b} = \frac{\text{volume cylinder and clearance volume}}{\text{clearance volume}},$$

t = temperature of charge and neutrals after mixing.

Heating Value of the Mixture. — After the mixture is drawn into the cylinder, its pressure is less and its temperature higher than without the cylinder as shown in the preceding article. The ratio of the specific volume of the mixture after it has been drawn into the cylinder to the specific volume at standard temperature and pressure is

$$\frac{V_a}{V_0} = \frac{P_0}{P_a} \times \frac{T_a}{T_0} = \frac{14.7}{13.7} \times \frac{580}{522} = 1.075 \times 1.11 = 1.195,$$

when the suction pressure is 1 pound below the atmosphere, and the temperature of the mixture in the cylinder is 120° F.

If we assume "a" to be the air required per cubic foot of gas, and "x" the proportion of actual air to theoretical air so that $(x - 1)$ will be the percentage excess air, then the volume of cold mixture per cubic foot of gas will be $(xa + 1)$ and the volume after it is drawn into the cylinder will be $\frac{V_a}{V_0}(xa + 1)$. If H is the heating value per cubic foot of gas under standard conditions, the heating value of the mixture per cubic foot after it is drawn into the cylinder will be $\dfrac{H}{\frac{V_a}{V_0}(xa + 1)}$.

Piston Displacement Per Horse-power. — The formula for the heating value per cubic foot of suction displacement (heating value of the mixture after it is drawn into the cylinder) gives us a means of determining the volume of normal charge required per horse-power.

Of the heat contained in a cubic foot of suction displacement, only a part is transformed into work. If we call the efficiency E, then $\dfrac{EH}{\frac{V_a}{V_0}(xa + 1)}$ is the heat per cubic foot that is actually transformed into work. Since it is necessary to transform 42.42 B.t.u. per minute to generate 1 horse-power, the number of cubic feet of suction displacement per minute will be

$$S = \frac{\dfrac{42.42}{EH}}{\dfrac{V_a}{V_0}(xa + 1)}. \tag{1}$$

This formula, then, gives us the number of cubic feet which the piston must displace, **on its suction stroke,** per minute, in

order to generate 1 horse-power. The value of this formula depends largely on the assumed efficiency E and a proper assumption can be made for this value only after long experience. It is evident that the value of E will increase with the compression and the allowable compression will be different for different gases. In order to give some idea of what efficiency might be expected, Table V is given here.

TABLE V

EFFICIENCY OF GAS-ENGINES BASED ON INDICATED HORSE-POWER

Compression ratio r	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5
Compression, gauge	46	57	72	86	102	118	134	151	168	186
Thermal efficiency	.18	.20	.215	.23	.243	.255	.265	.271	.276	.280

The compression pressures are based on the assumption that the suction pressure is 13.2 pounds absolute and the exponent of the compression curve is 1.35.

Formula (1) in this article gives the suction displacement per indicated horse-power minute, or the minimum that is to be provided. In order to allow for a small overload without sacrificing efficiency, this must be increased. As an engine may be rated in terms of brake horse-power, we must also change the formula to allow for that contingency. In general we may allow for an overload of 15 per cent and also assume the mechanical efficiency to be 85 per cent. We then have

$$S_1 = \frac{\dfrac{42.42}{EH}}{\dfrac{V_a}{V_0}(xa+1)} \text{ based on maximum I.H.P.,} \tag{2}$$

$$S_2 = 1.15\frac{\dfrac{42.42}{EH}}{\dfrac{V_a}{V_0}(xa+1)} \text{ based on rated I.H.P.,} \tag{3}$$

$$S_3 = 1.17\frac{\dfrac{42.42}{EH}}{\dfrac{V_a}{V_0}(xa+1)} \text{ based on maximum B.H.P.,} \tag{4}$$

$$S_4 = 1.35\frac{\dfrac{42.42}{EH}}{\dfrac{V_a}{V_0}(xa+1)} \text{ based on rated B.H.P.} \tag{5}$$

After the suction displacement has been determined, the size of the cylinder may be found when the type of engine is known. If the engine is a four-cycle, single-cylinder, the number of suction strokes will be $\frac{N}{2}$, where N is the revolutions per minute. If the horse-power HP be given as maximum indicated, then

$$HP \times \frac{\frac{42.42}{EH}}{\frac{V_a}{V_0}(xa+1)} = \frac{\pi D^2 L}{4} \times \frac{N}{2}, \tag{6}$$

where D is the diameter of the cylinder in feet,
L is the length of the stroke in feet,
N is the number of revolutions per minute.

Petroleum Fuels. — Petroleum is found in large quantities in Pennsylvania, Ohio, West Virginia, California, Texas and Russia, and to some extent in the East Indies. As it comes from the ground it is a mixture of many different hydrocarbons, with some sulphur and other impurities.

In the process of refining the light oils are driven off by distillation at comparatively low temperatures. The first vapor given off is called petroleum ether and is only a fractional per cent of the whole. This ether is all driven off below 140 degrees. Next come gasoline, the naphthas, kerosene and lubricating oils in order. After distillation there is left about 18 per cent of paraffine wax and residuum.

TABLE VI

AVERAGE ANALYSES OF AMERICAN FUEL OILS

	Gasoline.	Kerosene.	Crude oil.	Fuel oil.
Baumé degrees	70.2°	47°	18.3°	....
Specific gravity	0.704	0.863	0.877	0.939
Per cent carbon	82.50	84.00	84.24	83.04
Per cent hydrogen	14.90	14.14	13.44	11.58
Calorific value per pound (calculated) High	21,270	20,892	20,162	19,180
Calorific value per pound (calculated) Low	19,820	19,600	19,160	18,180
Calorific value per pound (actual test) High	19,740	20,849	19,560	19,121
Calorific value per pound (actual test) Low	18,500	18,755	18,636	....
Theoretical air, cu. ft. per lb.	193.5	189	187	178
Per cent nitrogen	1.93	1.84	0.2	0.60
Per cent oxygen	0.31	2.0	1.93	2.82
Per cent sulphur	0.09	0.23	0.823	0.90
Boiling point, degs. F.	167	437	....	....
Flash point, degs. F.	55	130	....	....

By using the cracking process, or destructive distillation, a considerable quantity of oils of a composition between kerosene and lubricating oil, is converted into hydrocarbons of lower density and boiling point, and thus made suitable for fuel and illuminating purposes.

The fuels that are light enough and have vapor pressures high enough to form an explosive mixture with air at atmospheric temperatures compose about 15 per cent of the petroleum. The demand for motor fuels, caused by the increase in the number of motor cars during the past ten years, has made it necessary to resort to the cracking process and also to devise some means of using the heavier fuels in automobile motors.

Gasoline as a Fuel. — The average gasoline found today in the markets is from 62 to 68° Baumé or 0.73 to 0.71 specific gravity. The average composition is about as follows:

Carbon,	83.5	per cent
Hydrogen,	15.5	per cent
Nitrogen, Oxygen, Sulphur, etc.	1.0	per cent
	100.0	

The Smithsonian Physical Tables give the heating value of 0.71 – 0.73 specific gravity gasoline as 19,980 – 20,520. This, of course, is the higher heating value; the lower would range from 18,580 to 19,120 B.t.u. per pound. The weight of air required per pound would be

$$\text{Carbon,} \quad 0.835 \times \frac{2.67}{0.23} = 9.68$$

$$\text{Hydrogen,} \quad 0.155 \times \frac{8}{0.23} = 5.40$$

$$\text{Total} \quad \overline{15.08} \text{ pounds.}$$

At 62 degrees and atmospheric pressure this air would have a volume of 15.08 × 13.14 = 198 cubic feet.

The volume of one pound of gasoline vapor is about 4.2 cubic feet at 62° F. and atmospheric pressure, and according to the analysis this requires 15 pounds or 198 cubic feet of air for theoretical combustion. If we allow some excess air, say 15 per cent, as we should with all fuels, the total air per pound of gasoline will be 17.25 pounds.

From Table IV it will appear that the saturation pressure for automobile gasoline, specific gravity 0.703, is 251 millimeters of

mercury at 86° F. The pressure of the atmosphere is, on the other hand, about twice this pressure, but the gasoline vapor is about three times as heavy as air. Therefore, while a saturated mixture of gasoline vapor and air contains twice as much air as gasoline, by volume, the gasoline is 50 per cent heavier than the air. Such a mixture is too rich to explode and must be diluted with at least ten times as much air by volume as the saturated mixture contains before it becomes explosive. On account of the high vapor tension of gasoline at the higher atmospheric temperatures, there is no trouble in forming an explosive mixture, since mixtures of air and gasoline in the ratio of 17 to 1 by weight may remain saturated at temperatures below 32° F., and a very simple carbureter will therefore suffice. However, although the atmospheric temperature is high, say 86 degrees, the gasoline in the carbureter will not be vaporized at that temperature but at one much lower. The latent heat of gasoline is about 180 B.t.u., and this heat must be supplied by the 18.25 pounds of mixture resulting from the vaporization of 1 pound of gasoline with 15 per cent excess of air. Hence, each pound of mixture must supply about 10 B.t.u. of the 180 required, and this will lower the temperature to about 46 degrees, as the specific heat of the mixture will be in the neighborhood of 0.25.

If the air is at 62 degrees, the temperature of the charge after vaporization, but before it goes into the cylinder, would be about 22 degrees, if no heat is supplied from an outside source. However, carbureters are usually jacketed with hot water, when the engine is water cooled, and the air for vaporization is taken from a jacket surrounding the hot exhaust pipe. Thus the temperature of the mixture may be 60 degrees when it leaves the carbureter. As it is drawn into the engine, it will be heated up by the hot valves and cylinder walls to a temperature of about 125 degrees. The ratio, then, of the specific volumes of the mixture after the completed suction stroke and its specific volume at standard conditions will be

$$\frac{V_a}{V_0} = \frac{P_0 T_a}{P_a T_0} = \frac{14.7}{13.2} \times \frac{585}{522} = 1.24, \tag{1}$$

assuming that the drop in pressure will be 1.5 pounds going into the cylinder, as it might easily be with a high-speed engine.

The estimate of the size of the cylinder per horse-power hour for this fuel will be as follows:

The volume of air required per cubic foot of the gasoline vapor will be

$$\frac{198}{4.2} = 47.2 \text{ cubic feet} \quad (a)$$

$$\begin{array}{rrl} \text{15 per cent excess} = & 7.1 & \text{cubic feet} \\ \text{Total} & \overline{54.3} & \text{cubic feet} \quad (xa) \end{array}$$

$$\frac{V_a}{V_0}(xa + 1) = 1.24\,(54.3 + 1) = 67.3 \text{ cubic feet.} \qquad (2)$$

Using the lower heating value of gasoline 18,580 the heating value per cubic foot of the vapor H will be

$$\frac{18{,}580}{4.2} = 4420 \text{ B.t.u.} \qquad (3)$$

The minimum suction displacement necessary per indicated horse-power per minute will be, if we use a compression ratio of 4 and an efficiency of 0.215 (Table V),

$$S_1 = \frac{\dfrac{42.42}{EH}}{\dfrac{V_a}{V_0}(xa + 1)} = \frac{\dfrac{42.42}{0.215 \times 4420}}{67.3} = 2.96 \text{ cu. ft.} \qquad (4)$$

If the engine to be designed is a single-cylinder, single-acting, four-cycle, there will be one suction stroke for each two revolutions. Then,

$$\frac{LAN}{2 \times IHP} = 2.96,$$

or

$$LA = \frac{2.96 \times 2 \times IHP}{N}, \qquad (5)$$

where L = stroke in feet, A = area piston in square feet, N = revolutions per minute. If the I.H.P. and N be known, the volume of the displacement LA may be solved for. Then an assumption of $\frac{L}{D}$, ratio of stroke to diameter, must be made. If $\frac{L}{D} = C$, then $L = CD$.

$$A = \frac{\pi D^2}{4}, \quad \text{and} \quad LA = \frac{C\pi D^3}{4}.$$

Substituting in (5) we have

$$D = \sqrt[3]{\frac{4 \times 2.96 \times 2 \times IHP}{\pi \times C \times N}}. \qquad (6)$$

Kerosene. — Like gasoline, kerosene is distilled from petroleum but is heavier than gasoline, has a higher flash point and lower vapor pressure of saturation for the same temperature. The specific gravity is from 0.79 to 0.82 or from 47 to 41° Baumé at 62 degrees.

The composition is about as follows:

$$\begin{aligned} C &= 0.845 \\ H_2 &= \underline{0.155} \\ & \ 1.000 \end{aligned}$$

The heating value is from 19,800 to 20,160 B.t.u. for the higher and 18,400 to 18,760 for the lower.

Since kerosene has a higher flash point and is less readily vaporized than gasoline, it is a safer fuel to use and to transport, but it does not make a convenient fuel for motor purposes. At ordinary temperatures a saturated mixture with air will not be explosive, so the fuel and preferably the air too must be heated to such a temperature that the process of vaporization will not bring the temperature of the mixture down below 80° F. This would result in a temperature of approximately 160° F. after the charge is drawn into the cylinder. If, as before, we assume the suction pressure to be 1.5 pounds below the atmosphere, we will have for the ratio of specific volumes

$$\frac{V_a}{V_0} = \frac{14.7}{13.2} \times \frac{620}{522} = 1.3. \qquad (1)$$

One pound of the fuel vapor occupies about 2.5 cubic feet at 62° F. and requires for combustion 190 cubic feet of air. The volume of air per cubic foot of vapor will be

	76 cu. ft.	(a)
Allowing 15 per cent excess	11.4 cu. ft.	
Total air per cu. ft. vapor	87.4 cu. ft.	(xa)

If we take 18,400 as the lower heating value, the heating value per cubic foot of vapor H will be

$$\frac{18{,}400}{2.5} = 7370 \text{ B.t.u.} \qquad (2)$$

The efficiency, with a ratio of compression of 4, will be 0.215. The suction displacement per maximum indicated horse-power per minute will be

$$S_1 = \frac{\dfrac{42.42}{EH}}{\dfrac{V_a}{V_0}(xa+1)} = \frac{\dfrac{42.42}{0.215 \times 7370}}{1.3\,(87.4+1)} = 3.08 \text{ cu. ft.} \qquad (3)$$

Assuming the same type of engine as in the preceding article, we find that the diameter of the cylinder in feet will be

$$D = \sqrt[3]{\frac{4 \times 3.08 \times 2 \times IHP}{\pi \times C \times N}}. \tag{4}$$

This shows that for the same power the engine using carbureted kerosene would have to be larger than an engine using carbureted gasoline by a ratio of $\frac{3.08}{2.96}$ for the same power.

Crude and Fuel Oil. — **Crude oil** is not used to any great extent for internal combustion engine fuel as it contains valuable light fuels which can be distilled off and sold for a much higher price than they would bring if sold in crude oil. **Fuel oil,** which is used in oil-engines, is heavier than crude, having a specific gravity of 0.92 to 0.94, whereas the specific gravity of crude oil is in the neighborhood of 0.88. Fuel oil is a little lower in carbon and hydrogen and correspondingly higher in nitrogen, oxygen and sulphur than the crude. In other words, fuel oil might be classed as the skim milk of petroleum. However, it is fairly good skim milk as the heating value is hardly 1000 B.t.u. less than the crude oil from which the fuel is obtained. Table VI shows the higher heating value of crude oil to be 20,162 — calculated, while the calorimeter gave a heating value of 19,560. These figures for the fuel oil are 19,180 and 19,121 B.t.u. respectively.

Crude, or fuel, oil is used in three ways in internal combustion engines. By the first method oil is forced into a hot chamber or vaporizer which forms part of the clearance space, during the compression stroke. This vaporizer is so arranged that it is not cleared of combustion products during the exhaust stroke, but the oil is vaporized by heat in an atmosphere of inert gases. As the compression pressure builds up, pure air is forced back into the vaporizer and when air enough to make an explosive mixture has been thus forced in, an explosion occurs. Low compression, 60 to 70 pounds per square inch, is used on this type of engine, so the thermal efficiency is low.

Another method of using heavy oil is to blow it into the clearance space, by means of high-pressure air, at the end of the compression stroke, and utilize a hot tube or pot to ignite the mixture thus formed. This hot tube retains heat from the previous explosion and there is no difficulty in maintaining a temperature far above the ignition temperature of the fuel. The compression

used is much higher than with a gas-engine, as it ranges from 150 to 300 pounds. The fuel is burned partially at constant volume and partially at constant pressure. The maximum pressure reached with 250 pounds compression pressure may be from 400 to 600 pounds, varying with the time of fuel injection. In tests on a two-cycle engine with compression at 260 pounds gauge, the author found maximum pressures reaching up to 580 pounds, with a mean effective pressure of 137 pounds. The fuel used was Pennsylvania fuel oil. This fuel was injected into the engine cylinder with the crank 10 degrees before dead center on the compression stroke.

A third method of using heavy oil is to inject it into the cylinder with high-pressure air, at the end of compression, as in the Diesel engine. Compression is usually carried to 500 pounds and the resulting temperature is sufficient to fire the oil without the use of an ignition system. The fuel is burned at constant pressure if the injection is regulated properly. If the injection is early, due to leaky valves or faulty setting, the maximum pressures reached run as high as 1000 pounds. Due to the high compression, efficiencies are very high. Tests have been reported where the efficiency reached 52 per cent, based on indicated horse-power. A more conservative figure would be from 42 to 45 per cent.

The theoretical air required per pound of crude oil is about 190 cubic feet, while for the average fuel oil, it is in the neighborhood of 180 cubic feet. The excess air should be greater in crude oil engines than in kerosene-, gasoline- or gas-engines. It should be at least 50 per cent in order to get perfect combustion.

The method used in determining the size of the cylinder for a given horse-power will be given in the design of the oil-engine.

Alcohol. — There are two kinds of alcohol that are available for engine fuel, ethyl, or grain alcohol (C_2H_6O), and methyl, or wood alcohol (CH_4O). Both these alcohols have a specific gravity of about 0.8 at 62° F. The boiling point of the former is 173 degrees and the latter 148 degrees F.

For the complete combustion of these fuels there is required:

	Oxygen per pound of fuel	Air per pound of fuel
C_2H_6O	2.086	9.03
CH_4O	1.500	6.50

The pure alcohols have heating values of about 16,000 B.t.u. per pound for the ethyl and 13,000 B.t.u. for the methyl. The manufactured alcohols never are pure but contain from 10 to 40

per cent water. With 14 per cent water the heating values would be about 13,700 and 11,000 B.t.u. respectively.

An internal revenue tax of two dollars and twenty cents per gallon on 100 per cent alcohol prevented the use of this fuel for the various arts and industries for which it is suited, until this tax law was repealed about 1906. Under the new law alcohol, which has been properly denatured by the addition of a definite amount of poisonous and repulsive ingredients to render it unfit for drinking, is exempt from the payment of any internal revenue tax. By law the "denatured alcohol" is to be made as follows: "to 100 parts of ethyl or grain alcohol by volume there must be added 10 volumes of methyl or wood alcohol and ½ volume of a heavy hydrocarbon, pyridine — denaturation benzol."

The latter ingredient is added for the purpose of giving the fuel a distinctive color and odor that can readily be recognized at inspection.

Another fuel, carbureted alcohol, is sometimes used. This is a mixture of the denatured alcohol with benzol, the latter being a purer article than that used for denaturing purposes. This benzol has a heating value a little lower than gasoline.

In France tests have been made with denatured alcohol and carbureted alcohol for motor use to determine their relative value. The analyses of the two fuels were as follows:

	Denatured alcohol.	Carbureted alcohol 50 per cent benzol.
Carbon.......	0.437	0.690
Hydrogen....	0.114	0.095
Oxygen.......	0.303	0.146
Water........	0.146	0.069
	1.000	1.000

Computed higher heating value.....9,938 B.t.u.....14,225 B.t.u.
Heating value by test............ 10,630 B.t.u.....14,180 B.t.u.

Some of the results obtained at these tests are given below, the figures being the relative fuel consumption by weight.

Power developed, B.H.P.	Denatured alcohol.	Carbureted alcohol 50 per cent benzol.
8.3	10	7.66
16.3	10	6.85
34.4	10	7.61

During 1906 the United States Department of Agriculture conducted a series of tests with alcohol as fuel in various types of engines to determine what difficulties would be encountered in using alcohol as a fuel for the types of engines already in use; to ascertain whether the engines could be made to operate satisfactorily on alcohol fuel, and what the consumption of alcohol would be compared with the fuel for which the engine was built; and finally, to throw light upon the action of alcohol as an engine fuel, so that some information might be obtained as to the nature and extent of modifications necessary in the engine mechanisms, in order that the most economical consumption of alcohol fuel might be obtained.

The following are some of the general conclusions reached as a result of the investigations:

1. Any gasoline-engine of the ordinary types can be run on alcohol fuel without any material change in the construction of the engine. The only difficulties likely to be encountered are in starting and in supplying a sufficient quantity of the fuel — a quantity which must be considerably greater than the quantity of gasoline required.

2. When an engine is run on alcohol its operation is less noisy, its maximum power is usually materially higher, and there is no danger of any injurious hammering such as may occur with gasoline.

3. Alcohol seems to be especially adopted as a fuel for automobile air-cooled engines, since the temperature of the engine may rise higher before auto-ignition takes place than is possible with gasoline fuel, and if premature ignition of alcohol does occur, no injurious hammering can result.

4. With any good small stationary engine as small a fuel consumption as 0.70 pound of gasoline or 1.16 pounds of alcohol per brake horse-power hour may reasonably be expected under favorable conditions. These values correspond to 0.118 and 0.170 gallon, respectively, or 0.95 pint of gasoline and 1.36 pints of alcohol. Based on the high calorific values of 21,120 B.t.u. per pound of gasoline and 11,880 per pound of alcohol, these consumptions represent thermal efficiencies of 17.2 per cent for the gasoline and 18.5 per cent for the alcohol.

Calculated on the basis of the low calorific value of 19,660 B.t.u. for the gasoline and 10,620 for the alcohol, the thermal efficien-

cies become 18.5 per cent for the gasoline and 20.7 for the alcohol. The ratio of consumptions is, alcohol to gasoline, 1.66 by weight, or 1.44 by volume.

Carburetion of Alcohol. — It is of special interest to determine the minimum temperature at which explosive mixtures of alcohol and air may be formed. This temperature may be determined by using Table VII, remembering that the ratio of pressures of the air and fuel vapor in a saturated mixture is equal to the ratio of the volumes of the two gases present.

TABLE VII

VAPOR PRESSURES OF ALCOHOL IN MILLIMETERS OF MERCURY

Temperature		Denatured alcohol.	Carbureted alcohol.	Temperature		Denatured alcohol.	Carbureted alcohol
°C.	°F.			°C.	°F.		
0	32	15	43	16	60.8	39.5	90
1	33.8	16	45	17	62.6	42	93.5
2	35.6	17	48	18	64.4	45	97.5
3	37.4	18	50	19	66.2	48	102
4	39.2	19	52.5	20	68.9	51	106.5
5	41	20	55	21	69.8	54	111.5
6	42.8	21.5	57.5	22	71.6	57	117.5
7	44.6	23	60.5	23	73.4	61	123.5
8	46.4	24.5	63	24	75.2	64	130
9	48.2	26	66.5	25	77	68	136.6
10	50	27.5	70	26	78.8	72	145
11	51.8	29	72.5	27	80.6	76.5	....
12	53.6	31	75.5	28	82.4	81	....
13	55.4	33	79	29	84.2	86	....
14	57.2	35	82.5	30	86	92	....
15	59	37	86	.....	....	....	....

If we assume that vapors have densities theoretically due to their molecular weight, we may apply a close approximation of the Laws of Mariotte and Gay-Lussac.

If V = the volume of air in cubic feet required for the combustion of 1 pound of fuel at 32 degrees and 760 millimeters,
t = temperature of the mixture of air and vapor, above 32° F.,
x = the vapor pressure corresponding to that temperature,
d = the density of the vapor at 32 degrees and 760 millimeters,
a = coefficient of expansion, by heat of the air and vapor.

Then $$V(1+at)\frac{760}{760-x} = \frac{1}{d}\,\frac{760}{x}\,\frac{(1+at)}{1},$$

since the volumes of air and vapor are equal in the mixture, or

$$\frac{V}{760 - x} = \frac{1}{dx}$$

and

$$x = \frac{760}{1 + dV}. \qquad (1)$$

Practice has shown that the excess air should be 50 per cent for alcohol instead of 15 per cent. Thus, denatured alcohol requires, by analysis, 7 to 8 pounds of air per pound of fuel, but experience has taught us to use 11.7 pounds. For 50 per cent carbureted alcohol the figure is 15.9 pounds, against 10.6 pounds by analysis.

Using formula (1) we find that the vapor pressure of the denatured alcohol with 50 per cent excess air is 48.6. For the 50 per cent carbureted alcohol with 50 per cent excess air, the vapor pressure is 23. Referring to Table VII we find that the lowest temperature at which the mixture can exist is 67° F. for the denatured and 12 degrees for the carbureted fuel.

If we consider the pressure of suction, say 13.2 pounds absolute, instead of atmospheric pressure, the above vapor pressures will become 43.7 and 20.7 millimeters respectively, and the temperatures of saturation 63° and 9° F.

If we require the fuel to be completely vaporized before it enters the cylinder, the heat of vaporization must be supplied by the air and fuel, which would correspond to using an unheated carbureter.

Taking denatured alcohol first we find that the latent heat at 62 degrees is 525 B.t.u. per pound. The specific heat of the gas is about 0.455 and of the air 0.24. The specific heat of the mixture will then be

$$\frac{0.24 \times 11.7 + 0.455}{12.7} = 0.257.$$

The fall in temperature of the mixture to supply the heat will then be

$$\frac{525}{0.257 \times 12.7} = 161° \text{ F.}$$

Since we must keep the mixture above 63 degrees, the initial temperature of the fuel and air will be

$$161 + 63 = 224° \text{ F.}$$

This shows the necessity of preheating the air and alcohol, and also why it is impossible to start an engine cold. If the 50 per

cent carbureted alcohol is used, calculations similar to those above show that the fuel and air must be heated only to 92° F. The specific heat of the carbureted mixture is 0.248 B.t.u. and the heat of vaporization is 350 B.t.u. The weight of mixture, per pound of fuel, is 16.9 pounds, and the temperature of saturation is 9° F.

It has been found by experiment that it is not necessary to heat the fuel and air as high as the preceding equations would appear to indicate. If the fuel is only partially vaporized in the carbureter, the vaporization may be completed by the hot walls of the engine and the hot neutrals, with marked success.

Since alcohols vaporize more slowly than gasoline, ports and valves should be large, to allow low velocities and thus have the air and fuel remain in contact for an appreciable time.

EXERCISES

1. Find the value of $\frac{V_a}{V_0}$ when the temperature in the cylinder at the end of suction is 107° F. and the pressure is 1.5 lbs. below the atmosphere.

2. Find the heating value per cubic foot of suction displacement of a gas whose heating value per cubic foot at standard condition is 135 B.t.u., air per cubic foot of gas = 1 cu. ft., $\frac{V_a}{V_0} = 1.3$. Allow 20 per cent excess of air.

3. Find the suction displacement per maximum I.H.P. per minute for the conditions given in Prob. 2. Assume compression of 160 lbs. gauge.

4. Find the size of the cylinder for an engine of 75 rated B.H.P., fuel to be natural gas of 980 B.t.u. per cu. ft. Standard conditions. Assume 10 cu. ft. of air per cu. ft. of gas, and allow 15 per cent excess. Compression to be 125 lbs. gauge. $\frac{V_a}{V_0} = 1.22$. Engine to be 4 cycle, 3 cylinder, single acting. Assume r.p.m.

5. Using the formula given in Chapter V for the heating value of liquid hydrocarbons, find the heating values of the kerosene and gasoline given in Table VI.

6. Find S_1 for denatured alcohol when

$$\begin{aligned}
\text{B.t.u. per pound} &= 10{,}450,\\
\text{Theoretical air} &= 102.5 \text{ cu. ft. per lb.},\\
\text{Volume 1 pound vapor} &= 9.4 \text{ cu. ft.},\\
\text{Excess air} &= 50 \text{ per cent},\\
T_a &= 63° \text{ F.},\\
P_a &= 684 \text{ mm. mercury, absolute},\\
\text{Compression} &= 186 \text{ lbs. gauge.}
\end{aligned}$$

CHAPTER VII

GASEOUS FUELS

Natural Gas — Anthracite Producer Gas — Bituminous Producer Gas — Illuminating Gas — Coke-oven Gas — Blast-furnace Gas

General. — The gases employed as fuels in internal combustion engines are of three distinct classes. The first is natural gas; the second, manufactured gas — that is, gas that is manufactured especially for use in making power or light; and the third class includes all by-product gases.

The first class includes only one and that is the "natural" gas which is used so extensively in some parts of the world, notably in western Pennsylvania and New York, West Virginia, Ohio, Indiana and some other parts of the United States. The second includes illuminating gas, either water or bench gas, and producer gas made from anthracite or bituminous coal or oil. The third includes principally coke-oven and blast-furnace gas.

In order to give some idea of the constituents of the various gaseous fuels Table VIII has been prepared and inserted here. The heating values, high and low, were calculated for the most part by using the heating values of the constituent gases as given in Table I. The air required was calculated by using Table II. The last column, Heating Value per Cubic Foot of Normal Charge, was calculated using 15 per cent excess air and a value of $\frac{V_a}{V_0}$ of 1.15, assuming the suction pressure to be 13.7 pounds absolute, and the temperature 120° F.

Natural Gas. — This gas is the most convenient of any for use in internal combustion engines. It requires no cleaning, it usually contains no sulphuric or other acid-forming constituents, and is very rich in heat-producing material. The supply is being exhausted rapidly, however, and in many localities where it is used, it is not to be relied upon when the demand is very great. The gas is usually sold at a price that makes it available for power purposes.

In finding the suction displacement necessary per horse-power hour, let us take the gas in the first line of Table VIII. The heating value per cubic foot of normal charge is 61.1 B.t.u. If we assume a compression ratio of 5.5, giving a compression pressure of 118 pounds, approximately, the efficiency expected will be 0.255. Then

$$S_1 = \frac{42.42}{0.255 \times 61.1} = 2.72 \text{ cubic feet per maximum I.H.P.}$$

Let us assume that we are to find the size of a cylinder for a four-cycle natural gas engine running 175 revolutions per minute, maximum indicated horse-power to be 75. We will assume that the ratio of length of stroke to diameter of cylinder will be 1.25 : 1. Then $L = 1.25\,D$. The volume of the cylinder will be

$$\frac{\pi D^2 L}{4} = \frac{1.25\,\pi D^3}{4}. \tag{1}$$

Since the engine is to be four-cycle, the suction takes place $\frac{N}{2}$ or $\frac{175}{2}$ times per minute. The total volume drawn in is then

$$\frac{175}{2} \times \frac{1.25\,\pi D^3}{4} = 86\,D^3. \tag{2}$$

Since this must equal the volume required to give the 75 horse-power, we have

$$86\,D^3 = 75 \times 2.72, \tag{3}$$

$$D = \sqrt[3]{\frac{75 \times 2.72}{86}} = 1.34 \text{ feet.} \tag{4}$$

Hence the diameter of the cylinder will be 16.1 inches and the length of the stroke 20.14 inches. (Needless to say the actual dimensions of the cylinder would be made 16 by 20.)

The mean effective pressure in this cylinder may be found from the formula

$$\text{m.e.p.} = \frac{I.H.P. \times 2 \times 33{,}000}{LAN}, \tag{5}$$

where N = number of revolutions per minute,
L = length of stroke in feet,
A = area of piston in square inches.

Substituting the proper values in this formula we get

$$\text{m.e.p.} = \frac{75 \times 2 \times 33{,}000}{\frac{20.14}{12} \times 204 \times 175} = 84.2. \tag{6}$$

TABLE VIII

Data on Common Fuel Gases

Kind of gas.	Constituents of gas in per cent volume.							Heating value of gas B.t.u. per cu. ft. at 62° F.		Cu. ft. of air required per cu. ft. of gas. Theoretical.	Heating value per cu. ft. of normal charge. B.t.u. = $\frac{H}{\frac{V_a}{V_0}(xs+1)}$
	H_2	CO	CH_4	C_2H_4	O_2	CO_2	N_2	High.	Low.		
Natural Gas:											
Olean, New York	....	0.50	96.5	1.00	2.00	....	....	998	896	9.33	61.1
Fostoria, Ohio	2.04	0.55	92.84	0.20	0.35	0.20	3.82	954	855	9.00	62.8
Pittsburgh, Pa.	26.16	0.80	65.25	6.30	0.80	0.60	....	852	763	7.50	66.2
Caspian Region, Russia	....	....	92.24	4.26	....	3.50	....	1004	904	9.40	63.7
Pechelbroun, Germany	....	3.50	77.40	4.80	1.80	3.60	8.90	874	788	8.21	62.8
Producer Gas:											
Anthracite coal	20.0	25.0	....	....	0.50	5.00	49.5	147	136	1.12	49.5
Bituminous coal	10.0	23.0	3.0	0.5	0.50	5.00	58.0	146	137	1.20	48.0
Coke	10.0	29.0	....	....	0.50	4.50	56.0	127	122	0.98	47.7
Illuminating Gas:											
Blue water gas	50.00	43.25	0.50	....	....	3.00	3.25	312	283	2.29	64.7
Carbureted water gas	40.00	19.00	25.00	8.50	0.50	3.00	4.00	583	528	5.03	64.8
Coal or bench gas	46.00	6.00	40.00	5.00	0.50	0.50	2.00	656	585	5.80	63.3
Oil gas	32.00	....	48.00	16.50	0.50	....	3.00	859	775	7.74	65.2
London, England	47.90	3.70	38.60	4.50	0.30	....	5.00	633	562	5.59	61.5
Coke-oven gas	53.00	6.00	35.00	2.00	....	2.00	2.00	580	514	5.06	62.8
Coke-oven gas	50.00	6.00	36.00	4.00	0.50	1.50	2.00	613	546	5.38	63.1
Blast-furnace gas	2.96	25.84	0.54	....	....	9.37	56.00	99	97	0 76	43.2
Blast-furnace gas	5.20	26.80	1.60	....	0.20	8.20	58.00	120	114	0.92	45.0

Now, in general, since the work done, in foot pounds, is equal to the pressure per square foot multiplied by the volume displaced, then the work done per horse-power per minute will be

$$144 \times \text{m.e.p.} \times S_1. \tag{7}$$

Since this must equal 33,000, we have, for the case under consideration,

$$144 \times \text{m.e.p.} \times 2.72 = 33{,}000 \tag{8}$$

or

$$\text{m.e.p.} = \frac{33{,}000}{144 \times 2.72} = 84.3. \tag{9}$$

This method may be used to check the first one.

Anthracite Producer Gas. — Anthracite producer gas, as shown by the analysis in Table VIII, requires 1.12 cubic feet of air per cubic foot of gas for perfect combustion. The heating value of the gas per cubic foot at 62° F. is 136 B.t.u. The heating value per cubic foot of suction displacement is 49.5 B.t.u. under the conditions stated in the table. If we allow a compression of 135 pounds, the efficiency, according to Table V, will be 0.265. Then the suction displacement per maximum indicated horsepower per minute will be

$$S_1 = \frac{42.42}{0.265 \times 49.5} = 3.24 \text{ cubic feet.}$$

Using the same data as in the last article, the diameter of the cylinder will be

$$D = \sqrt[3]{\frac{75 \times 3.24}{86}} = 1.41\dot{6} \text{ feet.}$$

The diameter in inches will be 17 and the length of the stroke will be 21.3.

The mean effective pressure will be

$$\text{m.e.p.} = \frac{75 \times 2 \times 33{,}000}{\frac{21.3}{12} \times 175 \times 227} = 70.3 \text{ pounds.}$$

Using the checking formula we get

$$\frac{33{,}000}{144 \times 3.24} = 70.6 \text{ pounds.}$$

Bituminous Producer Gas. — The bituminous producer gas given in Table VIII has a low heating value of 137 B.t.u. at 62° F. The air required is 1.2 cubic feet per cubic foot of gas,

theoretical. This is more than is required for the anthracite producer gas on account of the presence of methane and ethylene, neither of which is to be found in the anthracite gas.

The efficiency for this engine may be assumed to be the same as for the anthracite gas engine, namely, 0.265. Then the suction displacement per maximum indicated horse-power per minute will be

$$S_1 = \frac{42.42}{0.265 \times 48} = 3.33 \text{ cubic feet.}$$

The diameter of the cylinder required for the same 75 horse-power will be

$$D = \sqrt[3]{\frac{75 \times 3.33}{86}} = 1.43 \text{ feet.}$$

The diameter in inches will be 17.2 and the stroke 21.5. The mean effective pressure will be

$$\text{m.e.p.} = \frac{75 \times 33{,}000 \times 2}{21.5 \times 175 \times 232} = 68 \text{ pounds.}$$

Using the checking formula we get

$$\text{m.e.p.} = \frac{33{,}000}{144 \times 3.33} = 68.8 \text{ pounds.}$$

Illuminating Gas. — This gas is used only for small engines, or for engines that run infrequently. The gas is clean, high in heating value and always ready for use but the cost is prohibitive. The price is seldom below 80 cents per thousand cubic feet and that means very expensive fuel. It is used to some extent in emergency high-pressure pumping stations for fire service. Usually the engines are of the multicylinder, vertical type, and are geared to triplex plunger pumps. The Coney Island, N.Y., high-pressure pumping station contains three 175 horse-power Nash vertical three-cylinder engines geared to Gould triplex pumps. The engines have cylinders 13¾ inches in diameter and a stroke of 16 inches. They make 260 revolutions per minute. The gas has a heating value of 590 B.t.u. per cubic foot, and the engines use 17.4 cubic feet of gas per brake horse-power hour when running at rated capacity. These engines received their first severe test during a fire early in 1911. The first engine was put in service 28 seconds after the signal was received, and all three engines were in service within three minutes after the alarm was turned in. One engine ran steadily for a period of 58 hours, an-

other ran for 23 hours without a stop and the third was run as needed. The station, during that time, pumped 9,000,000 gallons of water and consumed 186,870 cubic feet of gas.

The pumping station shown in Fig. 23 is one of two in the city of Philadelphia used for high-pressure fire service. The two stations contain 20 illuminating gas engines aggregating 5350 brake horse-power. These engines, which are geared to triplex pumps, were manufactured by the Westinghouse Machine Company.

Fig. 23.

The Nash engines at Coney Island develop

$$\frac{175}{3 \times .85} = 68.5 \text{ indicated horse-power per cylinder.}$$

The ratio of stroke to diameter is

$$\frac{16}{13.75} = 1.162.$$

The gas described as coal or bench gas in Table VIII corresponds closely with the Coney Island gas. The heating value per cubic foot of suction displacement is 63.3 B.t.u. If we assume a compression of about 110 pounds gauge, the efficiency from Table V will be 0.25. For the suction displacement per indicated horse-power per minute, we will have

$$\frac{42.42}{63.3 \times 0.25} = 2.68 \text{ cubic feet.} \qquad (1)$$

To find the cylinder diameter, we equate the actual displacement on the suction strokes per minute to the required displacement, or

$$\frac{\pi D^2 L}{4} \times \frac{260}{2} = 70 \times 2.68. \tag{2}$$

But $$L = 1.162\,D.$$

Then $$\frac{\pi D^3}{4} \times 1.162 \times \frac{260}{2} = 70 \times 2.68, \tag{3}$$

$$D = \sqrt[3]{\frac{68.5 \times 2.68 \times 4}{\pi \times 1.162 \times 130}} = 1.158 \text{ feet.} \tag{4}$$

The diameter in inches will then be

$$1.158 \times 12 = 13.85 \tag{5}$$

which checks very closely the diameter given, which is 13$\frac{3}{4}$ inches.

Coke-oven Gas. — Up to about 1900 all the coke produced in the United States was made in the old "beehive" form of ovens. In these ovens the volatile matter that is distilled off is burned to keep the oven hot and to furnish heat for the coking. Since that time many "by-product" ovens have been built. As the gas is driven off from these ovens, it is led away and cooled, then washed. In these two operations all the tar is taken from the gas. The tar which is thus recovered amounts to about 40 pounds per ton of coal coked.

After the tar is removed, the ammonia is taken from the gas in one of several ways. Usually the gases are passed through weak sulphuric acid which picks up the ammonia forming the sulphate.

After the sulphate has been recovered, about half the gas is taken back to the ovens where it is burned to furnish heat for coking. The other half is available for power purposes, heating or lighting. The gas available for these uses contains about 7 per cent of the heat of the coal. Thus, the heat contained in the by-product gas per short ton of coal would be

$$2000 \times 14{,}000 \times 0.07 = 2{,}560{,}000 \text{ B.t.u.}$$

If the efficiency of a gas-engine be taken as 25 per cent, the power to be derived from this gas would be

$$\frac{2{,}560{,}000 \times 0.25}{2545} = 251 \text{ horse-power hours.}$$

If this gas has a heating value of 550 B.t.u. per cubic foot the volume available for power per ton of coal coked would be 4650

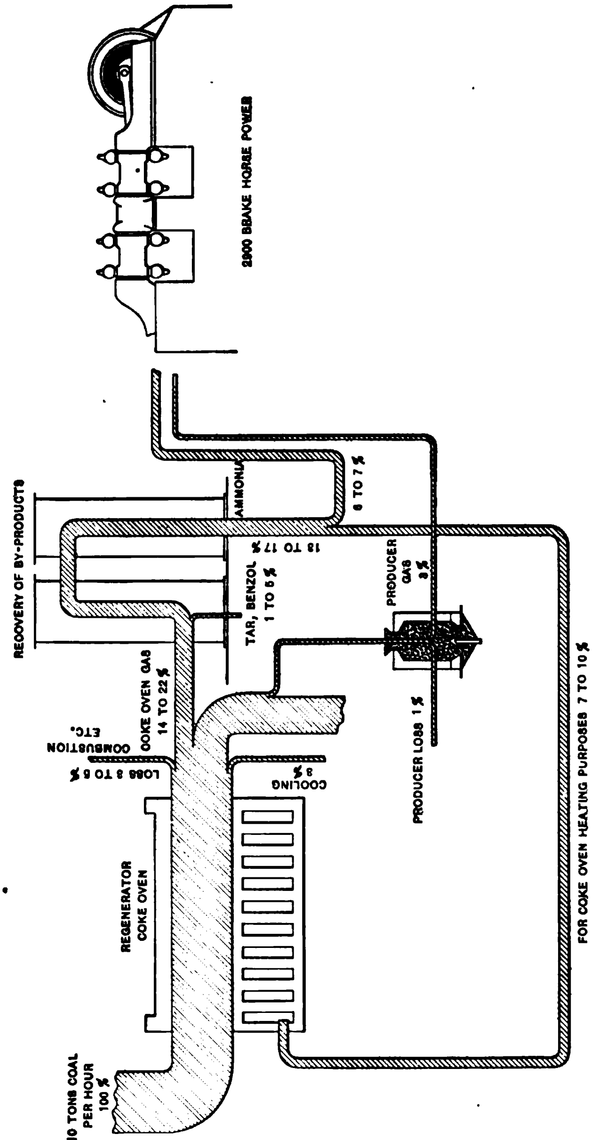


Fig. 24.—Heat Balance Diagram of a By-product Coke-oven.

cubic feet. Fig. 24 shows the heat balance of a battery of coke ovens of a capacity of 500 tons per day of 24 hours. The by-

product gas, if used in gas-engines, would generate 2000 brake horse-power continuously. If the coke "smalls" (refuse coke) were used in a gas-producer, and this gas mixed with the coke-oven gas, an additional 900 horse-power could be generated.

The coke-oven gas, Table VIII, line 14, shows a heating value of 514 B.t.u. per cubic foot at 62° F., with a heating value per cubic foot of normal charge of 62.8 B.t.u. If we assume a compression pressure of 120 pounds, giving us a ratio of compression

FIG. 25.—De La Vergne-Koerting Gas Blowing Engines. Lackawanna Steel Co., Buffalo, N. Y.

of 5.5, the probable efficiency will be 0.255. The suction displacement per maximum indicated horse-power per minute will be

$$S_1 = \frac{42.42}{0.255 \times 62.8} = 2.65 \text{ cubic feet.}$$

Assuming the same data used for natural gas, we find the diameter of the cylinder to be

$$D = \sqrt[3]{\frac{75 \times 2.65}{86}} = 1.325 \text{ feet.}$$

The diameter in inches will be $1.312 \times 12 = 15.9$ and the stroke will be $1.25 \times 15.9 = 19.85$ inches.

The mean effective pressure will be

$$\text{m.e.p.} = \frac{75 \times 2 \times 33{,}000 \times 12}{19.85 \times 175 \times 198.6} = 86.1 \text{ pounds.}$$

Using the checking formula we get

$$\text{m.e.p.} = \frac{33{,}000}{144 \times 2.65} = 85.5 \text{ pounds.}$$

Blast-furnace Gas. — The first attempt to use blast-furnace gas in gas-engines on a large scale in the United States was in 1903 at the Lackawanna Steel Company, Buffalo, N. Y. This plant consisted of sixteen 2000 horse-power blowing engines and eight 1000 horse-power generating engines. All the engines are of the twin, two-cylinder, two-cycle horizontal type, built by the De La Vergne Machine Company of New York working on the Koerting patents.

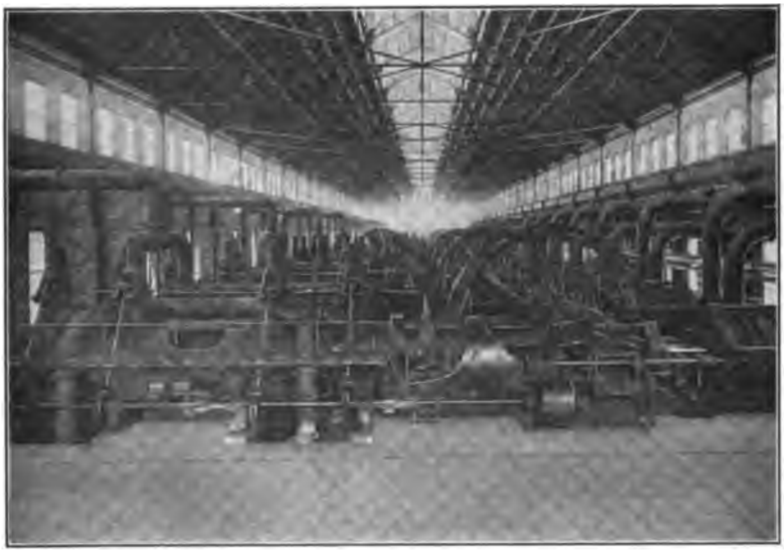


FIG. 26. — Westinghouse Gas Blowing Engines. Indiana Steel Co., Gary, Ind.

One of the houses containing eight of the 2000 horse-power blowing engines is shown in Fig. 25. While the gas-engines are horizontal, the blowing cylinders, of which there is one for each engine, are vertical, placed on *A* frames over the right-hand main shaft bearing. The blowing-cylinder connecting-rod is on the same crank-pin with the right-hand motor-cylinder rod.

These engines did not give entire satisfaction when first started, but they were subject to much adverse criticism that was entirely

unjust. As is the case in any new departure in the way of large power installations, it took some time to get these engines running satisfactorily, but the fact that they are giving better service in their eleventh year than they did in their first year of life is conclusive evidence that they are not failures by any means.

The largest installation of blast-furnace gas-engines in this country up to the present time is that of the Indiana Steel Company, Gary, Indiana. In this plant there have been installed 16 blowing-engines of the twin-tandem, four-cylinder, horizontal, four-cycle type. These engines have four power-cylinders each, 42 by 54 inches, and two blowing-cylinders, the capacity of the latter being 30,000 cubic feet of free air per minute. Eight of these engines were built by the Allis-Chalmers Company and eight by the Westinghouse Machine Company. Four of those built by the latter company are shown in Fig. 26.

In the central power station of the Gary plant there were originally installed 17 Allis-Chalmers engines of the same size as the blowing engines above. Fifteen of these engines are direct connected to 25-cycle, three-phase, 6600-volt generators and two are connected to 250-volt direct-current generators. The engines are rated at 4000 horse-power at 83⅓ revolutions per minute, the generator at 2000 kilowatts.

Since the original power engines were installed, six additional Allis-Chalmers engines have been installed. The later engines have a stroke of 60 inches instead of 54 inches, this being the only major change. Fig. 27 shows the power house containing the original Allis-Chalmers engines.

The standard blast-furnace, in the United States, has a capacity of 500 tons of iron every twenty-four hours. The heat balance of such a furnace is shown diagrammatically in Fig. 28. Of the 604 million heat units charged per hour in the form of coke 307 million are absorbed in the furnace proper, and 89 million in the stoves that heat the blast. Deducting 15 million for auxiliaries and washers 193 million are left for gas-engines. From this heat 19,200 brake horse-power could be generated and of that amount 3600 would be required for furnishing the blast; the balance, 15,600, would be available for general power purposes.

The most expensive part of the blast-furnace gas power installation, aside from the engines themselves, is the cleaning system. This system is elaborate, cleaning the gas by both dry and wet

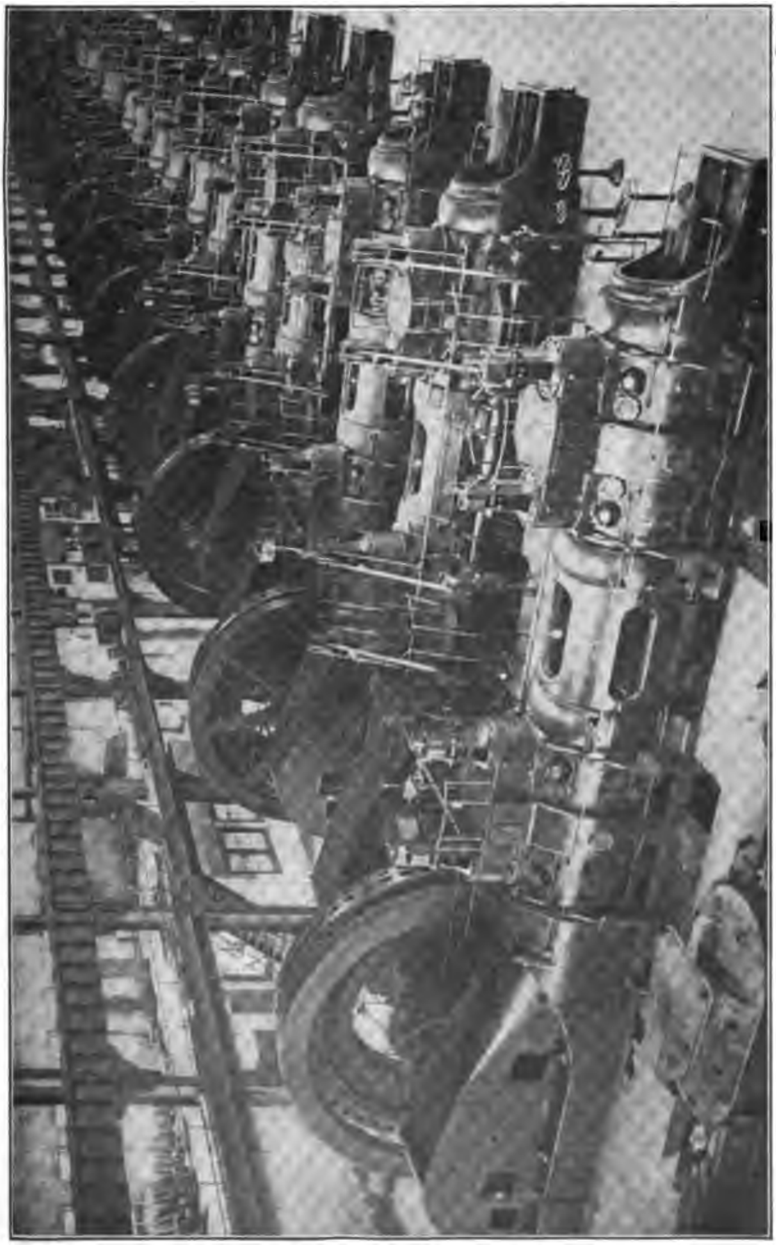


FIG. 27. — Allis-Chalmers Gas-engines Driving Electric Generators. Indiana Steel Co., Gary, Ind.

scrubbing. The Gary plant is up-to-date in this respect and a short description of that cleaning plant will not be out of place here.

Cleaning Blast-furnace Gas. — The gas is discharged from the top of the furnace through four outlets into two large pipes known as the "downcomers," which lead to a reservoir 30 feet in diameter by 40 feet high, called the "dry dust catcher." There a considerable part of the impurities settles to the bottom and the gas passes into another large pipe emptying into a supplemental tank 14 feet in diameter by 25 feet high, one of this size serving each pair of furnaces. This structure not only provides an additional dust catcher but acts also as a valve, being divided into two compartments partially filled with water. By increasing the height of the water in either one, the furnace on that side can be cut off as desired, and there will be no back-flow from the mains beyond. The two chambers of this tank discharge the gas into a ten-foot pipe, which carries the gas and the remaining impurities into the primary wet washers. There are three of these to each pair of furnaces, and each one has the capacity to take care of the gas from a single furnace, thus providing a spare washer for cleaning or repairing. These primary washers are cylindrical in form with conical top and bottom, and are about one-third full of water, an over-flow maintaining the proper level. Here the gas is discharged against the surface of the water from pipes with fluted ends, and then escapes around these ends into the openings of a larger main.

From the primary washers the gas is taken to the secondary washers, the first of this group being vertical scrubbers — drums 14 feet in diameter by 50 feet high. Water is sprayed from the tops of these towers, and the gas is let in near the bottom and rises up through the falling water. At the top the gas passes out and goes to the Thiessen washers, four of which are installed for each pair of blast furnaces. In these washers the gas is led between the wall of a cylinder and a revolving drum, the drum carrying a series of paddles or blades. A stream of water is spread into a film on the surface of the cylinder and the whirling drum throws such impurities as the gas still holds against the water film where they are caught and held. From these final washers the gas is conveyed under slight pressure to the two holders, each of which has a capacity of 200,000 cubic feet. From

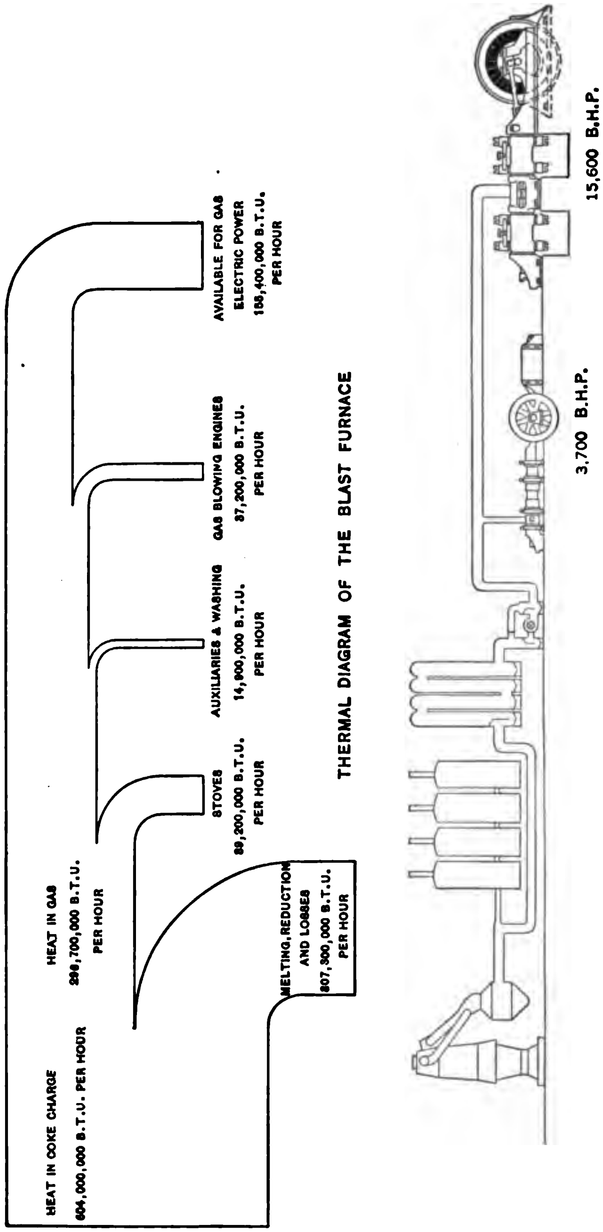


FIG. 28.—Heat Balance Diagram of a Blast Furnace.

these holders the gas is drawn, as required, by the electric power station and the blowing-engine houses.

Blast-furnace Gas as a Fuel. — Blast-furnace gas is low in heating value, varying from 95 to 105 B.t.u. per cubic foot. On very damp days in the summer, the moisture of the air is dissociated in the furnace and the hydrogen thus formed raises the heating value of the gas appreciably. The gas described in line 16, Table VIII, has a heating value of 97 B.t.u. at 62° F. It requires 0.76 cubic foot of air for theoretical combustion, resulting in a heating value per cubic foot of suction displacement (normal charge) of 43.2 B.t.u. The suction displacement per maximum indicated horse-power per minute would then be, assuming a compression of 150 pounds, giving an efficiency of 0.276,

$$S_1 = \frac{42.42}{0.276 \times 43.2} = 3.56 \text{ cubic feet.}$$

For the same 75 horse-power engine as used in the preceding examples the diameter of the cylinder would be, for blast-furnace gas,

$$D = \sqrt[3]{\frac{75 \times 3.56}{86}} = 1.46 \text{ feet.}$$

The diameter is $17\frac{1}{2}$ inches, and the stroke $1.25 \times 17\frac{1}{2} = 22$ inches.

For the mean effective pressure we have

$$\text{m.e.p.} = \frac{75 \times 2 \times 33{,}000}{\frac{22}{12} \times 175 \times 243} = 63.5 \text{ pounds.}$$

Using the checking formula we get

$$\text{m.e.p.} = \frac{33{,}000}{144 \times 3.56} = 64.4 \text{ pounds.}$$

EXERCISES

1. Check the higher heating values of the gases in Table VIII.
2. Check the lower heating values of the gases in Table VIII.
3. Check the cubic feet of air required for the gases in Table VIII.
4. Check the heating values per cubic foot of normal charge for the gases in Table VIII.
5. Determine the dimensions of a 125 *BHP* three-cylinder, single-acting gas-engine to use the oil gas, Table VIII. Ratio stroke to diameter, 1.25 to 1, r.p.m. 225, $\frac{V_a}{V_0} = 1.22$. Allow 20 per cent excess air. Compression, 110 lbs. g.

CHAPTER VIII

THE OTTO AND DIESEL CYCLES IN PRACTICE

Comparison of the Practical Points of These Two Principal Engine Cycles

Combustion. — The steps which take place during the process of adding heat to the charge in a gas engine cylinder are first, ignition; second, inflammation; third, explosion; fourth, combustion. **Ignition** takes place when sufficient heat is communicated to a mixture of gas and air by a flame, electric spark or hot tube to raise a particle of gas to the ignition temperature. Only a small portion of the charge need be heated in this way. **Inflammation** is the subsequent spreading of the flame throughout the mixture, or its propagation from one particle to another, until the whole volume is alight. **Explosion** follows when the mixture is completely inflamed and the maximum pressure attained. When all the gas in the cylinder is thoroughly lighted, the particles are driven widely apart, and thus the moment of complete inflammation will be that of maximum pressure. Complete inflammation and explosion are thus practically simultaneous. **Combustion** is complete when all the chemical changes have taken place and the gases have been transformed into water vapor and carbon dioxide. This moment may not, and usually does not, coincide with the point of explosion in the actual gas engine cylinder. The complete combustion is almost always delayed and it is this delay which gives the expansion line a form other than the true adiabatic for air or for the gas under question. This delay is appreciable and may be measured by methods which may be used without difficulty in the laboratory.

Flame Propagation. — The rate of flame propagation in a perfectly vaporized explosive mixture of correct proportions is very high, and ordinarily there is no advantage in increasing this velocity. In diluted mixtures, however, the rate of propagation is much lower and in many cases so low as to have a deleterious effect on the efficiency of the engine. Usually this low rate of propagation is responsible for the low efficiency of gas-engines at

light loads, when equipped with certain kinds of governors. Quite frequently special arrangements are made to increase the rate under these conditions.

Experiments by Mr. Grover at Yorkshire College show that if the excess air is replaced in part by burned gases, the rapidity of combustion may be increased. These experiments were made with coal gas. Experimenters are not united, however, as to the effect on the rapidity of combustion of replacing the excess air by neutrals in explosive mixtures in general. With some gases it seems to be a detriment to cut down the excess air by introducing neutrals.

Mr. Grover's experiments establish the following additional conclusions with respect to a coal gas mixture:

The highest pressures are obtained when the volume of air is slightly in excess of the amount required for complete combustion.

When the volume of the products of combustion does not exceed 58 per cent of the total mixture, the mixture is still explosive, provided the volume of air is not less than 5½ times the volume of the gas.

Higher pressures are recorded when the neutral gases take the place of excess air.

The above conclusions, with regard to the experiments, amount practically to saying that a perfect coal gas and air mixture may be diluted with burned gas amounting to 140 per cent of its volume and still be explosive; and the velocity of the flame propagation of such a mixture is more rapid than if air were used for a diluent.

The temperature of the charge when ignition takes place also has an influence on the rate of flame propagation, so that the higher the temperature, the more rapid the combustion becomes.

Experience also shows that mixtures in which the fuel is imperfectly vaporized often give unsatisfactory results due to slow combustion, and that a charge too weak to explode with a normal spark can be made to explode with a heavy spark.

Clerk's Experiments. — Mr. Dugald Clerk, in his experiments, considered that, to understand the action of gas in a cylinder, it was necessary to determine not only the time required to obtain a maximum pressure but also the duration of that pressure.

Mr. Clerk's experiments were with a small cylinder without

piston, filled with different explosive mixtures, connected to an indicator, the drum and paper of which were made to revolve so that each tenth of a revolution occupied 0.033 second. The pressure of the explosive gases forced up the indicator pencil, and by dividing the area of the moving drum on which the pencil traced curves into sections, the time occupied by the explosion and the cooling or reduction of pressure of the gases could be estimated within $\frac{1}{100}$ of a second. On this diagram the ordinates represented pressures, and the abscissæ the time of the explosion in fractions of a second. The maximum explosion pressure was developed in a closed vessel and therefore at constant volume, and the cylinder having no piston, no heat was expended in doing work. The diagrams showed that the pressure of the gases fell more slowly than it rose. The maximum pressure was produced in 0.026 second after ignition; the fall to atmospheric pressure and temperature occupied 1.5 seconds, or nearly sixty times as long. Without previous compression of the gases, the highest pressure obtainable with a dilution of 5 parts air to 1 part gas was only 96 pounds per square inch; with compression and a much weaker mixture this pressure was nearly doubled. The critical mixture, or the weakest dilution of gas and air that will ignite, varied according to the quality of the gas used. With Oldham gas a charge of 15 parts of air to 1 part of gas ignited, and the pressure was 40 pounds per square inch above the atmosphere. With Glasgow gas the critical mixture was 14 parts of air to 1 part of gas, and the pressure produced was 52 pounds per square inch.

The results of these experiments are shown in Table IX.

TABLE IX

Ratio by volume.		Maximum observed pressure. Lbs. per sq. in.	Time to reach maximum pressure. Seconds.
Gas.	Air.		
1	14	40.0	0.45
1	13	51.5	0.31
1	12	60.0	0.24
1	11	61.0	0.17
1	9	78.0	0.08
1	7	87.0	0.06
1	6	90.0	0.04
1	5	91.0	0.055
1	4	80.0	0.16

Massachusetts Institute Experiments. — Trials were carried out in 1898 at the Engineering Laboratory of the Massachusetts Institute of Technology to determine the interval of time in tenths of a second elapsing between ignition and the attainment of maximum pressure, and also the pressure of explosive mixtures of lighting gas and air in different proportions at each twelfth of a second. The pressures were shown on indicator diagrams, while a marker attached to a tuning fork traced simultaneously a time wave line of vibrations equal to one-sixtieth of a second. The explosions took place in a cast-iron cylinder, without a piston, of 310 cubic inches volume, to which a mercurial gauge for recording pressures, an air and a vacuum pump, and two electric batteries for firing the charge and keeping the tuning fork in vibration were connected. The pressure-recording apparatus consisted of an indicator paper fixed on a circular revolving disc driven at a uniform speed, on which by a special arrangement the tuning fork traced a succession of contiguous lines. Thus the time and pressure were recorded simultaneously, the atmospheric line and the diagrams of pressure both being circular.

TABLE X

RESULTS OF TESTS ON EXPLOSIVE MIXTURES OF ILLUMINATING GAS AND AIR

Mixture parts by volume, gas, air.	Maximum pressure, lbs. per sq. in.	Time of explosion, seconds.	Area, sq. in.	First one-fifth second.		
				Mean pressure, lbs. per sq. in.	Mean pressure ÷ proportion of gas.	Final pressure.
1-3	45	0.49	0.32	11	44	26
1-4	86	0.08	1.77	59	295	61
1-5	96	0.05	1.86	62	372	52
1-6	88	0.05	1.80	60	420	54
1-7	86	0.06	1.97	66	528	58
1-8	87	0.06	1.71	57	513	53
1-9	77	0.08	1.60	53	530	57
1-10	71	0.11	1.36	45	495	56
1-11	68	0.14	1.21	40	480	60
1-12	39	0.33	0.35	12	156	29
1-13	32	0.42	0.18	6	84	16
1-14	9	0.42	0.05	2	30	4

To determine the pressures the cylinder was first thoroughly cleansed, then connected to the vacuum pump, and the air exhausted to about one-sixth of an atmosphere. Into the partial

vacuum thus formed, giving a pressure of 5 inches of mercury, gas was admitted and the pressure was raised to atmospheric, thus producing a mixture of one part of gas to five of air. The charge was then fired electrically, the tuning fork set in vibration and the diagrams of pressure were taken. The proportion of gas and air could be varied by the vacuum in the pump.

The results of these tests are shown in Table X and graphically in Fig. 29.

It will be noticed by the diagrams in Fig. 29 that, generally speaking, the more rapid the combustion and the higher the

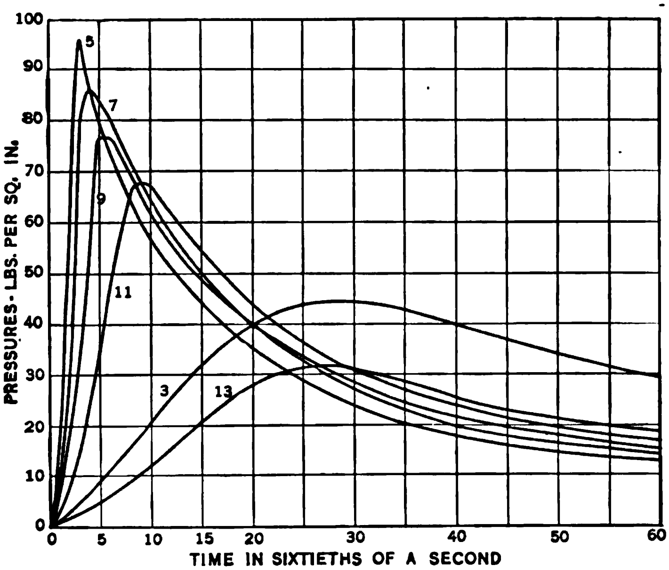


FIG. 29.

pressure, the lower the pressure line becomes after an interval of 1 second, except in the case of the very rich mixture, curve 3. The end of the first one-fifth second from the time of ignition is marked in the figure by a heavy line at the point $\frac{12}{60}$ of the total length of the diagrams, and at that line the final pressures of the first one-fifth second are measured. The areas obtained for the first one-fifth second and the mean pressures, columns 4 and 5 of Table X, refer also to the areas of the diagrams up to that line. The figures of the sixth column represent the efficiencies of combustion for each of the mixtures.

Mixtures Diluted with Combustion Products. — It might appear from Mr. Grover's experiments with mixtures diluted with air and burned gases that the neutrals in a weak mixture would tend to facilitate rather than retard the rate of combustion. However, the conditions under which these experiments were carried out differ materially from the conditions found in actual gas engine cylinders, and experience, with some fuels at least, show results differing from those found by Mr. Grover.

Assume a case of a throttling engine with a fairly high compression, which, due to faulty valve setting, does not relieve itself of burned gases readily. It is found that such an engine will give evidence of pre-ignitions on heavy loads, which are due probably to the excess heat which the neutrals give to the fresh charge. Under a light load, however, the operation of the engine is liable to be troubled with back firing into the mixing chamber or inlet valve casing, at the time of opening of the inlet valve for the admission of a fresh charge. This feature might be explained on the ground that an excessive amount of combustion products in a weak mixture makes it slow burning, not only during the expansion stroke but also during the exhaust stroke and up to the time of opening of the inlet valve for the following suction stroke.

After readjusting the valve setting, by opening the exhaust valve early and keeping it open as long as practical, an appreciable change will have been effected. The cylinder will become scavenged from combustion products as thoroughly as possible; hence the weak mixture on the same light load as before will be diluted by more air instead of by an excess of combustion products, and the back firing into the mixing chamber will have been cured. This tends to show that of the two mixtures of similar heating value, the latter, containing less combustion products, acts, in the working cylinder, as if it were the more inflammable.

Timing the Ignition. — Cards with normal, early and late ignitions are shown in Fig. 30. Card *A* shows what might be called normal ignition, card *B* early and card *C*, late ignition. Card *A* is ideal, that is, it is a little too perfect for an actual card. Of course the shape of the card will vary somewhat with the quality of the fuel. With a lean gas, such as producer or blast-furnace gas, the actual firing of the spark will occur at some point considerably before the end of the compression stroke, to give a card similar to the one shown at *A*. This may readily be understood

if the results of the experiments on flame propagation are studied carefully. Those tests show that the maximum pressure is attained an appreciable time after the spark is fired. This accounts for the necessity of firing the spark early when a lean mixture is being used.

Card *B* shows a spark that would probably be too early except for a high-speed engine. Later on in this book will be shown the effect of an early spark at different speeds, in connection with the inertia of the reciprocating parts. In general it might be said that early ignition may, and should, be used with lean mixtures and high speeds. However, using an early ignition with lean mixtures should result in card *A* and not card *B* if the speed

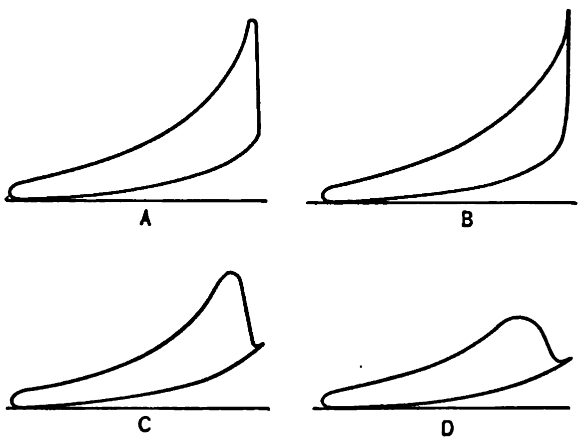


Fig. 30.

is not high. A card as shown at *B* is permissible only at high speed in order to get the cushioning effect.

Card *C* shows a diagram taken from an engine with late combustion. The cause might be any one of several, viz:—first, a late spark with normal mixture; second, a normal spark with a mixture that is too rich, as shown by curve 3, Fig. 29; third, a normal spark with a lean mixture, curve 13, Fig. 29; fourth, a cold engine; that is, when an engine is first started, even if the spark be set for normal operation, combustion would be late enough to give the card as shown at *C*.

This card *C* is indicative of low efficiency and should not be used where it can be avoided. The writer has found that in

certain engines which use a hot bulb for ignition, this card often results if too much water is injected into the cylinder. If the supply of water, or steam, where steam is used instead of water, is cut down to a minimum without producing premature explosions, much better efficiency will result and a larger load can be carried on the engine.

Suppression of Heat at Combustion. — The specific heats of gases have been found to be constant for a range of temperature up to about 400° F. Regnault has found that the specific heat at constant pressure c_p for air is 0.2375 from 0° to 200° C. Generally, both c_p and c_v for gases may be assumed to be constant up to 200° C. If we assume c_v constant throughout the range of temperature in a gas-engine, then the maximum temperature of combustion will be

$$T_c = T_b + \frac{Q}{c_v},$$

where T_c is the temperature of combustion and T_b is the temperature at the end of compression, or the beginning of combustion.

The formula would give much higher temperatures than are reached in an actual engine for several reasons. The specific heat of gas is not constant but, as a general rule, the law of variation is expressed by a linear equation; thus,

$$c_v = a + bt,$$
$$c_p = a' + bt.$$

Further, some of the heat generated by the combustion is dissipated into the cylinder walls. There has been a good deal of controversy as to the relative amount of the heat that is lost in this way, but indications seem to point to only a small loss. Probably an appreciable amount is lost through dissociation of the gases at temperatures reached in actual practice, and it is certain that the generation of heat is less than the theoretical owing to the fact that combustion does not take place at constant volume. The piston moves an appreciable amount during combustion and this movement increases the farther the mixture is from the theoretical explosive mixture. That is, with a mixture that is too rich or too lean, combustion takes place during the movement of the piston when the gas is doing work. The heat equivalent of this work, then, does not go to help raise the temperature, consequently the temperature reached is lower than the theoretical.

The values for specific heats of products of combustion, at constant pressure and constant volume, are shown in Table XI, the temperature t being expressed in degrees Fahrenheit. These values are those determined by Mallard and Le Chatelier.

TABLE XI

SPECIFIC HEAT AT CONSTANT PRESSURE

$$\begin{aligned} CO_2 &= 0.185 + 0.000093\,t, \\ H_2O &= 0.415 + 0.000202\,t, \\ N_2 &= 0.240 + 0.000024\,t, \\ O_2 &= 0.211 + 0.000021\,t, \\ \text{Air} &= 0.233 + 0.000023\,t. \end{aligned}$$

SPECIFIC HEAT AT CONSTANT VOLUME

$$\begin{aligned} CO_2 &= 0.140 + 0.000093\,t, \\ H_2O &= 0.360 + 0.000202\,t, \\ N_2 &= 0.173 + 0.000024\,t, \\ O_2 &= 0.150 + 0.000021\,t, \\ \text{Air} &= 0.165 + 0.000023\,t. \end{aligned}$$

To illustrate the method of finding the probable combustion pressure of a fuel in a gas-engine, the following example is worked out.

Let us assume a fuel, the composition of which is, by weight.

	Per cent
$H_2 =$	1.01
$CO =$	25.78
$CO_2 =$	17.39
$CH_4 =$	0.92
$N_2 =$	54.90
	100.00

The weight of air to burn a pound of gas will then be, from Table II,

$$\begin{aligned} H_2 &= 0.01 \times 34.640 = 0.3464 \\ CO &= 0.258 \times 2.477 = 0.6390 \\ CH_4 &= 0.009 \times 17.36 = \underline{0.1559} \\ & \qquad\qquad\qquad\quad 1.1413 \end{aligned}$$

The excess air will be

$$\begin{aligned} &0.15 \times 1.1413 = 0.1712 \text{ pounds.} \\ &0.1712 \times 0.23 = 0.0398 \text{ pounds excess oxygen,} \\ &0.1712 - 0.0398 = 0.1314 \text{ pounds excess nitrogen.} \end{aligned}$$

The weights of the combustion products will then be, from Table II:

	CO_2	H_2O	N	O
H_2	....	0.09	0.2664	....
CH_4	0.024575	0.02025	0.1199	....
CO	0.40560	....	0.4915	....
CO_2	0.17400	....	....	....
N_2	....	....	0.5490	....
Excess	....	....	0.1314	0.0398
Totals per pound gas	0.60435	0.11025	1.5582	0.0398

The maximum temperature of combustion is probably in the neighborhood of 3000° F., as an average.

The absolute temperature after compression is, if we assume a compression ratio of 6, corresponding to 135 pounds per square inch compression,

$$T_b = r^{n-1} \times T_a = 6^{0.35}\,(460 + 80) = 1.87 \times 540 = 1010 \text{ degrees}$$

or 550° above Fahrenheit zero.

The mean temperature during combustion will be about 1800 degrees. The mean specific heat of the gases in the products of combustion may then be found from Table XI.

These are

$$\begin{aligned} CO_2 &= 0.317, \\ H_2O &= 0.690, \\ N_2 &= 0.218, \\ O_2 &= 0.190. \end{aligned}$$

The mean c_v for the products of combustion will then be found as follows:

$$\begin{aligned} CO_2 &= 0.60435 \times 0.317 = 0.1916 \\ H_2O &= 0.11025 \times 0.690 = 0.0761 \\ N_2 &= 1.55820 \times 0.218 = 0.3397 \\ O_2 &= \underline{0.0398} \times 0.190 = \underline{0.0076} \\ & \ 2.3126 \qquad\qquad\quad 0.6150 \end{aligned}$$

The C_v per pound will then be $\dfrac{0.6150}{2.3126} = 0.2659$.

In order to find the heating value per pound of mixture, including excess air, we may use the constituents given per pound of fuel. These must be reduced to percentages, and each fractional part in the mixture multiplied by its heating value.

	Gas and air, per lb. fuel.	Per cent weight of mixture.	B.t.u. per lb. lower value.
H_2....................	0.010	0.00432 ×	52,275 = 258
CO....................	0.258	0.11150 ×	4,380 = 488
CO_2....................	0.174	0.07519	
CH_4....................	0.009	0.00389 ×	21,544 = 84
N_2....................	0.549	0.23725	
Air....................	1.314	0.56786	
	2.314	1.00001	830

Thus, 830 is the heating value of the fresh charge. But the fresh charge is mixed with the neutrals before combustion takes place, hence the heating value will be lowered. The heating value of the charge after being mixed with the neutrals will be

$$830 \times \frac{r-1}{r} = 830 \times \frac{5}{6} = 691 \text{ B.t.u. per pound.}$$

Under the discussion of the Otto cycle, Chapter IV, the ratio of temperatures before and after combustion is given by the formula

$$\frac{T_c}{T_b} = \frac{Q}{c_v T_b} + 1.$$

Since the combustion occurs at constant volume, we may write

$$\frac{P_c}{P_b} = \frac{Q}{c_v T_b} + 1,$$

P_c being the pressure after and P_b before combustion. It has been shown that not all the heat of combustion is effective in increasing the pressure but some of it is lost in various ways. If we assume the loss to be 0.15 per cent, the available heat will be 0.85 Q. Then

$$\frac{P_c}{P_b} = \frac{0.85 \times 691}{0.2659 \times 1010} + 1 = 2.18 + 1 = 3.18,$$

$$P_c = 3.18 \times 150 = 477 \text{ pounds absolute.}$$

This result is probably not far from the pressure that would actually be attained in the engine cylinder with the kind of gas used in this example. The value of this formula, however, depends upon how accurately the percentage heat lost during combustion can be estimated. There is no reliable data or information at present on the subject, and it seems to the author that, until such data is available, this method should be used with caution. An-

other method which has been found to give satisfactory results will be discussed under the design of the engine.

The Diesel Cycle. — Dr. Diesel's original intention was to build an engine working on practically the constant temperature cycle, and though the Diesel engines work in a manner somewhat different from that which the inventor intended, it might be interesting and of value to know what the inventor tried to do in his early experiments. In the patent specification dated August 27, 1892, Dr. Diesel proposed to compress air alone to the highest temperature of the cycle,

"that is to say the temperature at which the subsequent combustion has to take place, namely, the combustion temperature (not the burning or ignition point.)"

"If it be desired, for instance, that the late combustion shall take place at a temperature of 700° C., the pressure will be 64 atmospheres; for 800° C., the pressure will be 90 atmospheres, and so on."

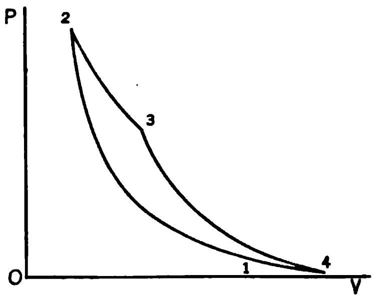


Fig. 31.

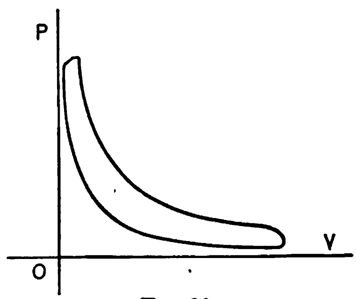


Fig. 32.

In this compressed body of air the fuel was to be injected and ignited, far above the temperature necessary for inflammation. Simultaneously with the gradual introduction of fuel, an expansion of the body of air was to take place, which was to be regulated in such a manner that the cooling caused by expansion would destroy at each moment the heat produced by the combustion of the several introduced particles of fuel; owing to this the combustion would show its effect, not by an increase of temperature but solely by work done. The diagram corresponding to this explanation is shown in Fig. 31.

Diesel found it impossible to carry out this cycle in practice owing to the high pressures involved, so he compromised on the card shown in Fig. 32. In this card, compression is carried to a

pressure that will result in a temperature high enough to ignite the fuel. This compression is in the neighborhood of 500 pounds, resulting in a temperature of 1000° F. in actual practice. The fuel is injected at a point and at a rate which will give constant-pressure combustion but not constant temperature, in fact the temperature rises until it is little, if any, lower than in the constant-volume engine. This temperature rise may continue for a short period after cut-off. This type of engine was described by Diesel in his second patent.

It is very important to note from a practical point of view that the duration of combustion is considerable and hence there is a longer time available for getting rid of the heat by the jacket water than in a gas-engine, where combustion is rapid and the temperature rises very rapidly. It follows, therefore, that for the same maximum pressures in the cylinder of a gas-engine and a Diesel engine the temperature rise in the first case would be higher than in the latter. Since a Diesel engine may work with much higher compression pressure than a gas-engine and yet not be subjected to higher temperatures, hence the fact already well established, that greater powers may be obtained per cylinder than with the ordinary types of internal combustion engines. There are, however, other factors entering into this matter.

EXERCISES

1. Find the c_p of the following gas at 2000° F.: CO_2 = 27 per cent, H_2O = 14 per cent, N_2 = 23 per cent, Air = Bal. by weight.

2. Find the c_v of the following gas at 1500° F.: CO_2 = 28 per cent, N_2 = 48 per cent, O_2 = 6 per cent, Air = Bal. by volume.

3. Given: H_2 = 2 per cent, CO = 25 per cent, CH_4 = 18 per cent, CO_2 = 12 per cent, N_2 = 43 per cent, by weight. Find the weights of the combustion products when the gas is burned with 20 per cent excess of air.

4. Find the c_p of the combustion products of the gas given in Prob. 3.

5. Find the B.t.u. per pound of the mixture of the gas given in Prob. 3 with air for combustion and 20 per cent excess.

6. Find the probable explosion pressure of gas in Prob. 3 when $r = 7$. Assume efficiency of combustion to be 80 per cent.

CHAPTER IX

IGNITION — TIMING — HOT-TUBE — MAKE-AND-BREAK — JUMP-SPARK — PARTS OF IGNITION SYSTEM

Timing the Ignition. — The ignition of any gas-engine should be properly timed. The time of ignition varies with conditions, and cannot be set by any fixed rule. If electric ignition is used, the device is so arranged that the point of ignition can be varied at will within certain limits. In general the conditions that affect the ignition are: first, speed of engine; second, kind of fuel; third, quality of mixture.

The higher the speed of the engine, the earlier the spark should be set. This is easily explained by the fact that it takes an appreciable time for combustion to occur after the spark is fired. At high speeds, then, the spark must be set early in order to have the maximum pressure occur at the beginning of the stroke. Manufacturers of automobiles sometimes make arrangements for the spark to be advanced or retarded automatically as the engine speeds up or slows down. This is an advantage as it relieves the driver of the care of the timing and it also gives more accurate results than could be obtained by hand manipulation. As an automobile engine slows down, due to the increase in load, the period of time in which combustion may take place becomes lengthened, and if the spark is set early for high speed, it occurs too soon for the reduced speed, and maximum pressure is set up before the piston reaches the dead center. This trouble makes itself known, first, by knocking in the cylinder, and finally by sudden stopping of the engine due to the early high pressure.

Fuels that naturally burn **rapidly** require a later spark than the slower-burning fuels. Thus, gasoline or natural gas engines should have later ignition than producer or blast-furnace gas-engines, if the compression is the same. But, since higher compression increases the rate of combustion, the higher the compression the later may be the ignition for a given fuel.

When an engine is running at a **light load** (with a lean mix-

ture), combustion will be slower and ignition should be correspondingly earlier. Likewise, a mixture that is too rich would require an early spark for the same reason. The theoretically proper mixtures have the most rapid combustion and require the latest spark.

Several systems of ignition are used at the present time, although the major part of all engines being made today have an electrical

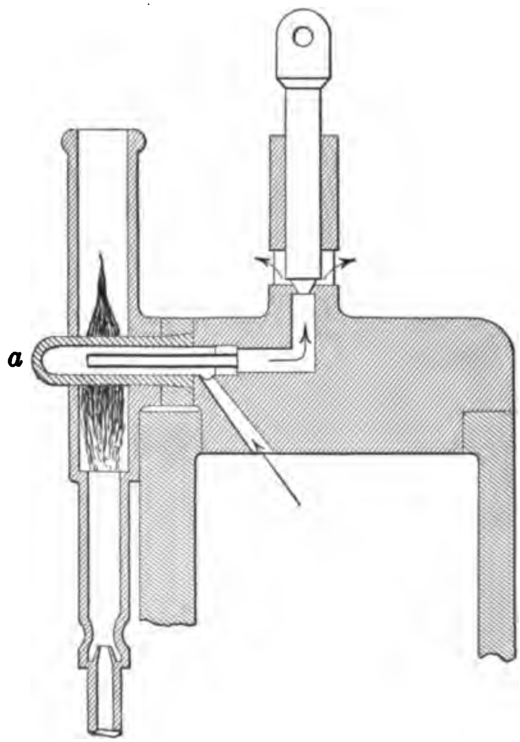


FIG. 33. — Typical Arrangement of Hot-tube

system. Others are the hot-tube and the system in which ignition is due to high temperature of compression.

The Hot-tube. — A typical hot-tube arrangement is shown in Fig. 33. The tube *a* is made of porcelain or a nickel alloy, and communicates with the combustion chamber. It is heated, for starting, by a Bunsen burner or a gasoline torch. After the engine has been running for a short time, the tube will remain at a temperature high enough to fire the charge.

When the exhaust stroke has been completed, the tube will remain full of inert gases which will be at approximately atmospheric pressure. As the pressure builds up during compression, a small portion of the fresh charge is forced into the tube until finally it comes in contact with the part of the tube that is at or above the ignition temperature of the mixture. When this occurs, the flame strikes back along the tube and ignites the whole charge.

The **timing** can never be exact when a hot-tube is used. If the system described above, where the tube retains heat from the former explosion, is used, the ignition point will vary with the compression or the quality of the fuel. But, if the Bunsen burner is used to keep the tube hot, timing may be changed to a slight extent by moving the burner out towards the end of the tube for later, or in towards the combustion chamber for earlier, ignition.

The **advantages** of the hot-tube ignition are that it has no moving parts to wear out or get out of order and it is simple and is certain. However, it has the **disadvantage** of being low in efficiency, and, as has already been mentioned, it cannot be regulated with any degree of accuracy.

Electric Ignition. — There are many forms of electric ignition, and the scope of this book will not permit more than a superficial examination of a few of the more important types. Generally, the electric ignition system consists of a **circuit of wire,** or a combination of wire and the metal of the machine, a gap or **circuit-breaking device** inside the cylinder, a **commutator** or **distributing device** outside the cylinder and a **source of electric current.** Other devices used are a **switch, induction-coil** or **kick-coil** and **condenser.** The source of current is the **wet-cell, dry-cell, storage-battery** or **magneto.**

The systems may be classified as **low-** and **high-tension,** or **jump-spark** and **make-and-break.** The make-and-break system is usually used when low-tension current is used, and the jump-spark system with high-tension current.

Low-tension Ignition.— As the name implies, low-voltage current is used with this system, the source of which may be dry-batteries, storage-cells, generator or magneto. The tension is, on the average, from 5 to **6 volts,** and the resistance is such that from 2 to 5 **amperes** flow when the circuit is closed.

In Fig. 34 is shown the general arrangement of a make-and-break low-tension ignition system. When the "electrodes" are in contact and the switch closed, the current flows through the circuit until the electrodes are separated. When they are suddenly separated, a spark passes between them, igniting the charge in the cylinder, if the charge is at the proper condition of mixture and compression. The passing of the spark is due to the action of the coil in the circuit. This coil is termed the "reactance," "kick" or "intensifying" coil. It consists of a number of turns of insulated wire wound on a soft-iron core. On account of the

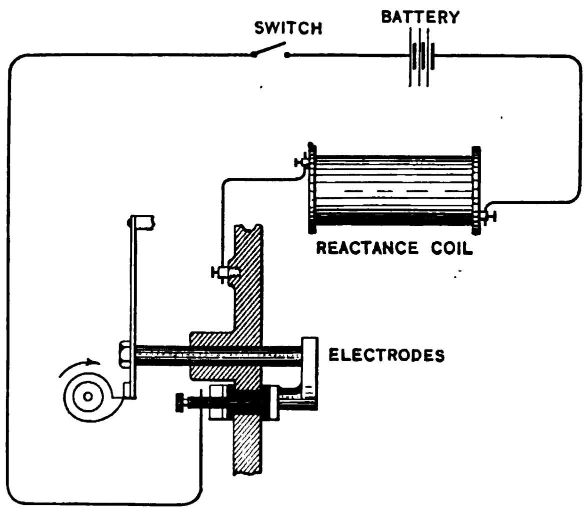


Fig. 34. — Diagram of Make-and-break Ignition System.

inductive action of such a coil a momentary increase of voltage is caused when the circuit is broken by the moving electrode.

This make-and-break apparatus is very simple electrically, but mechanically it is more complicated. The device used to give motion to the moving electrode must necessarily be more or less complicated in order to make it convenient to change the spark. It is also troublesome to prevent leakage around the moving electrode where it enters the cylinder. Lubrication is difficult and it is hard to keep the electrodes cool enough to work efficiently. For high-speed engines special refinements are necessary in order to make the inertia effects as small as possible, and to make the lost motion, due to wear, a minimum.

A typical **igniter** is shown in Fig. 35. The points on the electrodes are brought into contact some time before the spark is to be made. A push rod engages the "hammer trigger" which has no rigid connection with the moving electrode, being connected only through a spring. As the end of the push rod slips off the igniter cam, it allows the electrodes to be pulled apart suddenly by the spring *S*, making the spark in the cylinder.

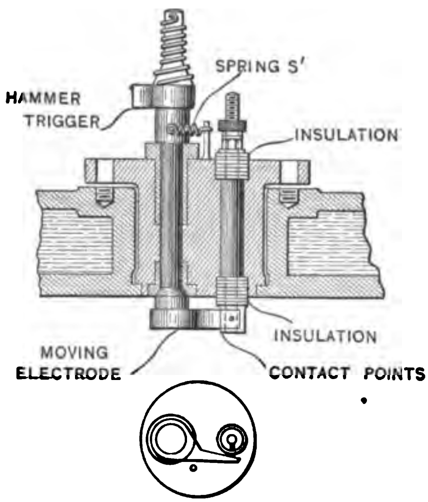


FIG. 35. — Make-and-break Igniter.

The make-and-break system described in the last paragraph is of the hammer type. Another type, the Wiper Contact Igniter, is shown in Fig. 36. This type is used by the Foos Gas Engine Company and is favored by them on the ground that the rubbing of one electrode over the other keeps both clean and free from carbon and other deposits, which may interfere, to a certain extent, with the proper working of the hammer type.

There are also a number of electrically or magnetically operated "make-and-break" plugs on the market, designed to make a self-contained mechanism, which need only to be wired to the source of the current and a timing device, eliminating push rods, cams and other numerous mechanical parts. This type of ignition is generally constructed in such a way that the electric current energizes an electromagnet which causes the motion of the electrode and also serves as a kick coil. One type of magnetic plug is made by the Bosch Magneto Company for operation in connection with certain of their magnetos. This plug is illustrated in Figs. 37a, 37b and 37c. The assembled plug is shown in Fig. 37a. *B* is the coil body with sleeve and terminal, shown in section in Fig. 37b. In Fig. 37c the part marked 1 is the interrupter, 2 is the pole piece and 3 is the U-shaped spring.

The interrupter lever rests on a steel knife edge and also on the

contact block. One end of spring 3 presses against the back of the interrupter just inside the knife edge. This arrangement facilitates the easy action of the interrupter and spring and reduces wear.

The spring has a very slight movement, and as it is embedded in the coil body, it is protected from the hot gases.

The lower end of the coil body *B* is screwed into the pole piece of the plug, thus forming a complete electromagnetic system, the armature of which is the upper end of the interrupter lever.

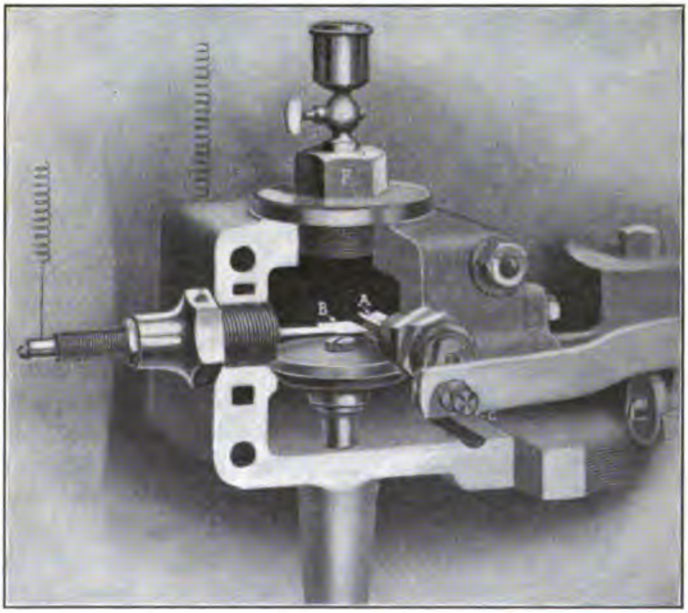


Fig. 36. — Foos "Wiper-Contact" Igniter.

The contact pieces 20 and 21 are mounted upon the interrupter lever and the plug body *A*, the latter reaching into the interior of the combustion chamber. The two contacts are pressed together by spring 3, and are separated only at the time of sparking, at which moment a current flows through the coil. For this purpose one end of the winding is connected to the current-carrying ring 6 and through this to the insulated terminal 9 whereas the other end is electrically connected by means of the screw 26 to the body of the coil. The body of the coil, as well as the pole

piece and the interrupter, are insulated from the plug body *A* by means of the steatite cone 22 and the washers 18.

Variation in the timing is effected by rotating the timing lever 12, by means of which the interruption is caused to occur earlier

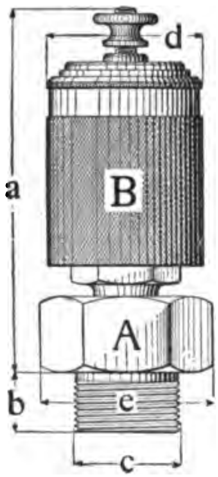


FIG. 37a.

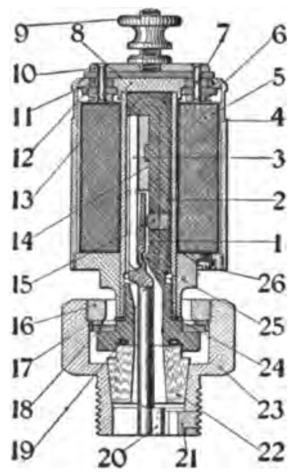


FIG. 37b.

Bosch Magnetic Igniter.

or later in the revolution of the armature; the spark in the magnetic plug occurs earlier or later. The timing of the ignition may be varied so as to occur at any time during the rotation through an angle of about 50 degrees on the armature shaft, which is equal to 60 degrees on the crank shaft of a three-cylinder motor, 50 degrees on a four-cylinder motor, about 34 degrees on a six-cylinder motor and 25 degrees on an eight-cylinder motor.

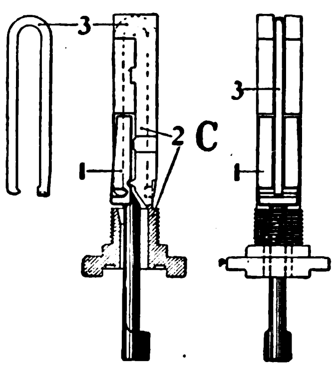


FIG. 37c.
Bosch Magnetic Igniter.

The magnetic plug ignition may be switched off by short circuiting the current permanently. This is done by connecting the short-circuiting terminal 22 through an insulated wire to the engine by means of a single-pole switch.

Make-and-break Mechanism. — One mechanical device used for moving the electrode is shown in Fig. 38. The cam *C* on

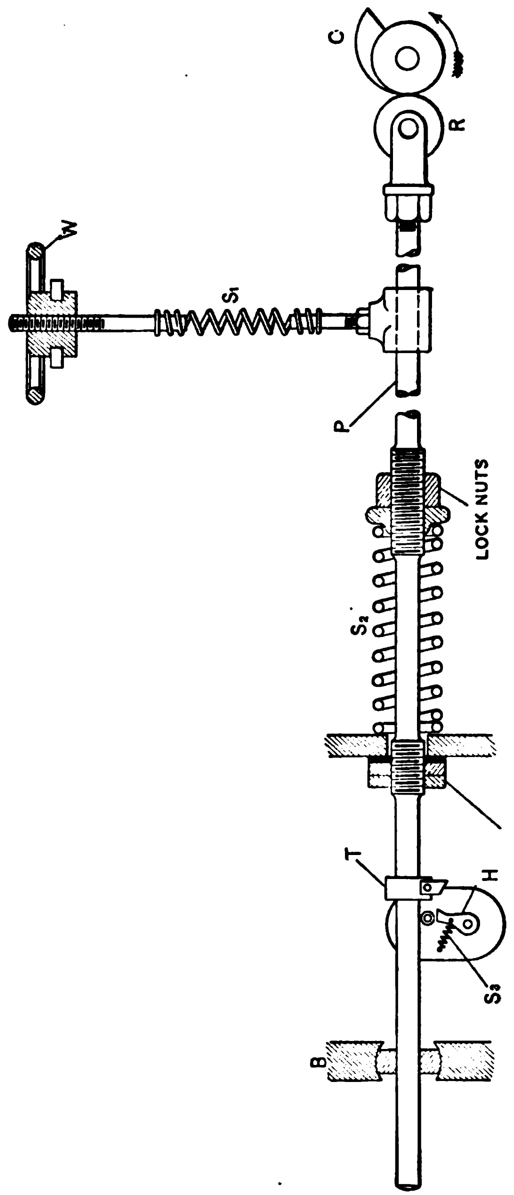


FIG. 38.—Make-and-break Mechanism.

the cam-shaft gradually moves the push rod *P* to the left. As this is done, the tappet *T*, which is hinged, passes over the hammer *H* without disturbing it. However, when the roller *R* drops off the high point of the cam, the spring *S* quickly forces the push rod to the right and the tappet now strikes *H*, forcing the electrodes apart in the combustion chamber, causing the spark. The current is timed so that it is passing through the electrodes only at the instant it is needed. As the electrodes are in contact a large percentage of the time, the current is shut off during this period.

In order to change the **time** of the ignition, the hand wheel *W* is supplied. It is evident that if the roller *R* be raised a certain distance, the spark will be earlier in proportion and if the roller be lowered, the ignition will be later. The wheel serves this purpose and to allow this motion the bearing *B* is pivoted.

The spring S_1 is inserted for flexibility. If a rigid rod were used in place of the spring, and the engine ran backwards for part of a revolution in stopping, as often occurs, the mechanism would be damaged. The spring prevents any damage as it allows the cam to force the roller down, extending the spring. As the roller is released, the spring brings the mechanism back to normal position.

High-tension Ignition. — A single electric spark, or series of sparks, jumping across a permanent break in the metallic circuit, and passing through the combustible mixture in the cylinder of an engine, is a means adopted to a very considerable extent for igniting the charge in an internal combustion engine. A high electromotive force, or tension, is required to force the spark across the gap. This is secured by use of the induction coil which changes a current of an ampere or two, with a tension of a few volts, into a current of much higher tension with a corresponding decrease in the current. The low-tension current is supplied by a dry-battery, storage-battery, magneto or small lighting generator. A "timer" is used in connection with the battery. The function of the timer is to close the battery circuit so that a current can flow from the battery through the induction coil at the proper instant to produce a spark in the combustion chamber.

The **generator**, the **timer**, the **induction-coil** and a **distributer** for directing the high-tension current to the different combustion chambers of a multicylinder motor are all sometimes brought

together and embodied in a single piece of apparatus. This combination is called the **high-tension magneto.**

Spark Plugs. — The jump-spark igniter is, in almost all cases, made up of a central metal wire surrounded by a thick tube of insulating material which, in turn, fits into a hollow metal plug threaded on the outside so as to screw into a threaded hole in the wall of the cylinder of the engine or into some part that fits into the cylinder wall. The insulating material is generally either **porcelain** or **mica.** The former is used in one tubular piece, and the latter is made up of numerous discs perforated for a central wire and placed side by side over it. **Lava** is used for insulation to a limited extent. When the porcelain is used, tight joints are made between the porcelain and the central wire and between the porcelain and the outer bushing by the use of **asbestos fibre packing** or **soft copper washers.** In general practice either the central wire of the plug terminates in a wire of smaller diameter brought near the outer shell, or a small wire is fastened to the outer shell and brought near the enlarged end of the central wire. The gap thus left in either case is called the "spark gap." Its width is usually about $\frac{1}{32}$ inch. The size of the threads of the American made plugs corresponds to the standard half-inch pipe tap. French plugs are smaller at the threaded part of the outer shell and approximately the same dimensions elsewhere.

FIG. 39.—Spark Plug for High-tension Ignition.

Timers for High-tension Ignition. — When a battery is used to supply current for the **high-tension** jump-spark ignition, a **timer** is placed in the battery circuit to close it at the instant a spark is required. The timer thus controls the time of flow of the low-tension current. One of the principal parts of the timer is stationary, except for a slight adjusting movement, and the other rotates. These are electrically insulated from each other. As the rotor revolves, a metal contact piece on it comes against the metal of the stationary part at intervals and closes the circuit. In the more common type, used where there is a separate induction-coil for each cylinder, there is a contact piece on the

stationary part for each coil. The contact pieces are placed around the circular path of the rotating piece in order to close the circuit at the time required by each cylinder of the motor. The stationary part is adjustable to a certain degree by rotary motion around the rotor shaft, so that the time of closing the circuit, and hence the explosions, can be varied to suit the needs of the motor.

In the best modern designs the stationary part generally consists of a ring of **wood fibre** supported on a metal part that is bored to receive the shaft of the rotor. The contact points of the stationary part are attached to the insulating ring so as to be insulated from each other and from the shaft of the rotor. The rotor has only one contact piece, and in some designs it has rigid metallic connection with the rotor shaft; in others it is insulated from the shaft, but is permanently connected by rubbing or rolling contact with the metal ring that is part of the stationary member of the timer, and is electrically connected to the metal of the motor. The contacts are pressed together by the action of a spring. The moving parts are packed in soft grease or lubricated with plenty of oil.

In automobile practice the frequent movement of the adjustable piece in advancing and retarding the spark, is liable to break the wires that lead from the contact pieces to other parts of the apparatus. In order to prevent this trouble the adjustable part is surrounded by a case that is truly stationary with regard to the motor frame, and the leading-out wires are connected to binding posts on the casing. The electrical connections between the case and the adjustable part are made by sliding contacts.

The speed of the rotor of the timer is half that of the crank-shaft in four-cycle engines and the same as the crank-shaft in two-cycle engines.

Induction-coils. — The induction-coil used for **high-tension** ignition in gas-engine and motor practice has a central core of very soft, small iron wires arranged in a bundle. Insulating material in the form of a tube covers the core. Comparatively coarse copper wire is wound around the insulating tube in the form of a solenoid coil of a few layers of wire with several turns in each layer. This is the "low-tension," "primary" or "battery" coil. The turns of wire are insulated from each other by using a wire with an insulating cover or by carefully winding bare wire with sheet insulation between the layers, and then filling the spaces

in between the wires with paraffine. One end of the primary coil is attached to a binding post for receiving a battery wire, and the other end connects to a device for interrupting the current.

The interrupter has a thin flat spring (vibrator, trembler) that is rigidly held at one end so that a metal contact point near the free end is pressed against the mating point. The metal parts to which the two contact points are attached are electrically insulated from each other when the points are separated. The second wire from the battery is connected to the part of the interrupter that is insulated from the side to which the primary-coil wire is connected. The free end of the spring has attached to it a disc of soft iron that is held just opposite one end of a soft-iron core of the coil and a short distance from it. When a current of electricity is passed through the coil it magnetizes the core which then attracts the metal disc and draws it, together with the free end of the spring, toward the core. The contact points are thus separated and the current interrupted. The core then quickly loses its magnetism, and the elasticity of the spring brings the points together again so that the current again passes through the coil. This operation is repeated and continued as long as sufficient current is supplied.

A second coil, the **secondary** or **high-tension,** is wound over the first. It is of exceedingly fine wire and has a great many turns. The turns and layers are separated from each other as in the primary coil. One end of the high-tension coil is connected to the same binding post as the end of the low-tension coil. The other end of the high-tension coil is attached to a binding post of its own. The apparatus thus has three binding posts for receiving wires from the outside. One terminal is at the battery side of the interrupter; another, which may be called the intermediate terminal, is between the ends of the high- and low-tension coils; and the third terminal is at the remaining end of the high-tension coil.

When the **battery current** stops flowing through the **primary winding** it induces a current of **high pressure** and small volume, of amperage, in the **secondary winding.** For ignition purposes the tension of the secondary current should be high enough to give a spark across a one-quarter-inch air gap.

An electric **condenser** is used in connection with an induction-coil. Its function is to strengthen the action of the coil and pre-

vent the contact points of the interrupter from fusing. When the contact points of the interrupter are separated there is a tendency for the battery current to keep flowing in an arc across the gap

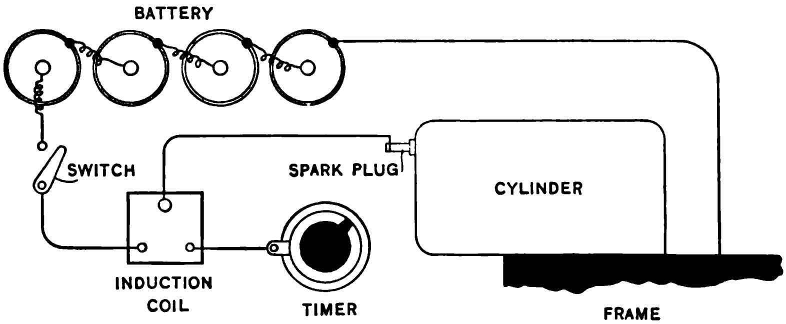


Fig. 40. — Wiring Diagram for High-tension System.

formed. The magnetic core, acting inductively on the primary coil, has a tendency to maintain the arc. The condenser counteracts the combined effort by receiving and storing the electrical energy and thus breaking down the arc quickly. The energy

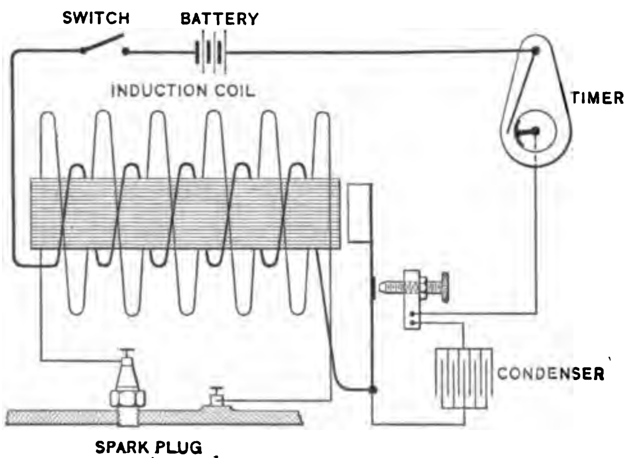


Fig. 41. — Diagram of High-tension Ignition System with Condenser.

stored in the condenser is probably discharged back through the primary circuit immediately after the primary circuit is broken, thus further increasing the inductive action and the strength of the spark.

The **condenser** is made up of sheets of tin foil and paraffined paper laid together alternately, the paper insulating the sheets of foil from each other. Alternate sheets of foil are connected together electrically to form one pole of the condenser, and the remaining sheets are likewise connected together to form the other pole. One of these poles is connected to the battery side of the interrupter, and the other to the primary coil side.

American made induction-coils for ignition purposes are generally wound to operate on 6 or 7 volts. Most of the European made coils are for a voltage on the primary of 4 volts. Higher voltages than these should not be used on account of trouble in melting contact points and breaking down the insulation of the winding.

The wires and connections of a high-tension ignition system are shown in Fig. 40. In this case the condenser and induction-coil are contained in one case and a minimum amount of wire is used, the wires being grounded to the metal of the engine wherever possible. In Fig. 41 the wiring is shown with a separate condenser connected across the vibrator points. Here the only "ground" is at the spark plug itself.

CHAPTER X

CARBURETION AND CARBURETERS — TYPES OF CARBURETERS

Carburetion. — Carburetion is the process of making a combustible mixture of fuel vapor and air by passing air, cold or hot, over or through a spray of the volatile liquid fuel. The amount of fuel vapor that will be carried by the air depends on the vapor tension of the fuel. When the vapor tension of the fuel is, under ordinary atmospheric conditions, much less than that of air, it may happen that the ratio of fuel to air in a saturated mixture is too small to form a good explosive mixture. On the other hand, fuels with high vapor pressures form saturated mixtures with air at ordinary temperatures and pressures that are too rich to burn and must be diluted before they will explode in a gas-engine cylinder.

In Chapter VI it was pointed out why certain liquid fuels form explosive mixtures with air very readily at normal temperatures and others must be heated before such mixtures can be formed. The **gasoline** which is used today in automobile motors and stationary gasoline-engines has so much lower vapor pressure compared to that used ten years ago that the matter of carburetion has become very important; and at the same time it is a problem that is very difficult to solve in a way that is entirely satisfactory. Formerly, carbureters were not water-jacketed, nor was any special provision made for supplying warm air to the carbureter. Today both these measures are resorted to in order to give good results.

When it is attempted to use **kerosene or alcohol,** the matter of preheating the fuel and air becomes still more important. With kerosene it is necessary to supply heat on account of its low vapor pressure of saturation at ordinary temperatures. With alcohol the same difficulty presents itself, and in addition, the water which is always present in commercial alcohol gives trouble.

Kerosene cannot, in general, be used with a gasoline carbureter even if provision is made for jacketing with hot water and supplying hot air. It is true that almost any gasoline-engine

which has been operated long enough to have reached a constant temperature can be made to continue in operation on a kerosene and air mixture formed in a gasoline carbureter. The operation will, however, be more or less uncertain and it will, in general, be only a matter of a few days before the interior of the combustion chamber is covered with a deposit of carbon and the piston rings are gummed tight in their grooves.

Kerosene, being less volatile than gasoline at ordinary temperatures and pressures, is not thoroughly vaporized in the ordinary gasoline carbureter, but some of the fuel is carried over into the cylinder in the liquid form. Part of this liquid will remain in that state until combustion has started. When this occurs the high temperature will tend to vaporize the excess liquid, but there is also another process known as "cracking" that goes on simultaneously. Cracking consists in breaking up the molecules of the liquid kerosene, some of the resultant products being finely divided carbon and heavy viscous or tarry liquids. It is this carbon and these heavy liquids that cause the trouble previously referred to by collecting in the cylinder.

The more the kerosene is broken up, that is, the more perfectly it is atomized or sprayed into the air, the less chance will there be for the collection of the liquid on cold surfaces and the more rapid will be the vaporization in the cylinder of the engine. Fine spraying is therefore advantageous, and any jet carbureter will work best when the pressure available for causing the flow is so great that the needle valve must be almost entirely closed.

It has also been found by experience that if a certain amount of **water vapor** or a finely divided water spray is mixed with the kerosene-air mixture, conditions are materially improved. No really satisfactory explanation of the full action of water vapor has yet been offered, but it seems probable that it is very complicated and of a physical as well as chemical nature.

The use of water makes possible the use of high compression without danger of pre-ignition and this makes possible the attainment of high efficiency. This is probably due to the fact that water absorbs heat during compression, thus keeping the temperature below that which would cause spontaneous ignition, and there may also be some sort of chemical action whereby the presence of the water prevents the breaking up of, or the combination of, certain of the constituents of the fuel to form compounds that have

very low ignition temperatures. The presence of water also tends to prevent the deposit of carbon in the cylinder and to make combustion more perfect, so that the exhaust is less smoky.

Types of Carbureters. — When a high grade of **gasoline** is used for fuel, a perfectly satisfactory mixture of air and vapor can be made very readily and at ordinary temperatures by simply injecting the liquid into the air as it passes on its way to the cylinder. Devices arranged thus to inject fuel into the air may be classified under the head of jet carbureters. They are almost the only kind used in this country at present, although very satisfactory carbureting devices of other types have been used and are being used.

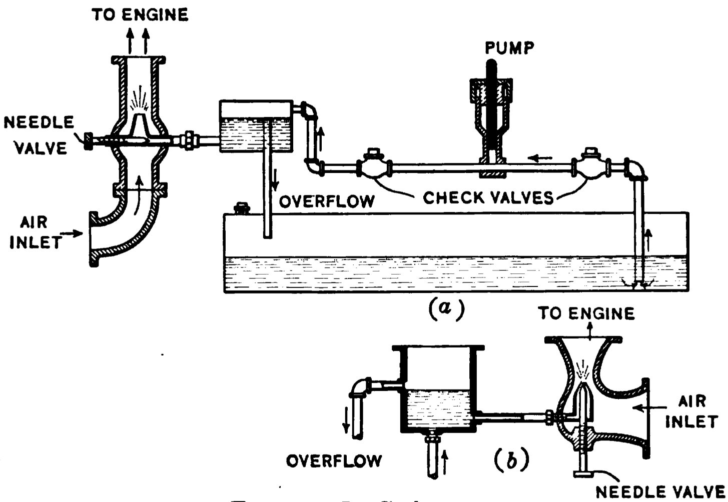


Fig. 42. — Jet Carbureters.

Jet Carbureters. — In this form of carbureter the nozzle is supplied from a small reservoir in which the fuel is maintained at a fixed level slightly below that of the nozzle so that normally there will be no flow from the nozzle. During the suction stroke of the engine a flow is established as described below. The level in the reservoir is maintained in several ways. One method of keeping the level constant is by means of a float and needle valve. This method has given rise to a particular type of carbureter known as the "float-feed." This type will be explained later. A simple method of maintaining a constant level is shown in Fig. 42. An overflow from the reservoir back to the fuel tank carries away any

surplus fuel. In this type a pump must be used to lift the fuel from the storage tank to the constant-level reservoir.

It will be noticed that in both (*a*) and (*b*), Fig. 42, the nozzle enters the air pipe leading to the inlet valve of the engine, or is centered in that pipe. The diameter of the suction or air pipe is often reduced as shown at (*b*), thus forming a Venturi tube. This increases the velocity of the inlet air at that point and assists in the mechanical separation of the particles of gasoline. During the suction stroke of the engine, the pressure within the air pipe is reduced below that of the atmosphere, and the air rushing past the nozzle creates an additional slight pressure drop immediately opposite the end of the latter. As a result of this double reduction in pressure opposite the end of the nozzle, a small jet of gasoline is discharged into the air current by atmospheric pressure on the surface of the liquid in the reservoir. With a Venturi passage a considerable decrease in pressure may be obtained at the nozzle and the pressure built up again to nearly atmospheric conditions at the inlet valve.

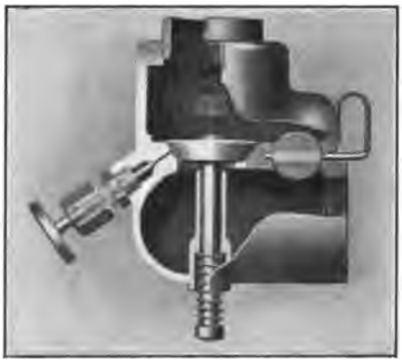


FIG. 43. — Carbureter Valve.

The quantity of gasoline thus discharged can be regulated by means of the needle valve shown in the figure. When hit-and-miss governing is used it is obvious that no fuel need be drawn out of the nozzle and wasted during the missed cycles. When throttling governing is used, the throttle valve may be placed on the engine side of the carbureter, thus controlling the quantity and velocity of the air through the latter and indirectly controlling the amount of gasoline discharged into that air.

Another simple form of carbureter, known as the carbureter valve, is shown in Fig. 43. The light disc acting as a poppet valve is inserted in the air pipe and closes off the supply of the air and the gasoline as shown. During the suction stroke this valve is raised by the atmospheric pressure, and, as the gasoline outlet is uncovered, a jet spurts into the air as it passes on its way to the cylinder. The gasoline may be supplied from a constant head

reservoir with a level slightly below the discharge point, or preferably under slight pressure, pressure to be from natural head or by furnished air pressure in the supply tank.

Carbureting valves do not form a very good mixture of fuel vapor and air and are consequently used only on small engines. They are used frequently on two-cycle engines having crank-case compression and are fairly successful as the churning action in the crank-case has a tendency to complete the thorough mixing started by the valve. The two properties which recommend this valve for use on small engines are low cost and simplicity.

Float Carbureters. — The carbureter used on practically all multi-cylinder gasoline-engines and on all automobile and marine motors

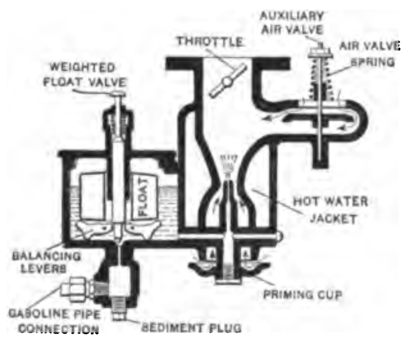


Fig. 44. — Carbureter with Eccentric Float.

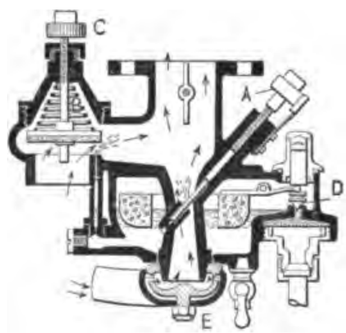


Fig. 45. — Carbureter with Concentric Float.

is of the jet type in which the level in the small reservoir near the jet is kept constant by means of a float and needle valve. The float is made of sheet copper or cork, the latter material being the favorite.

The position of the float relative to the spray jet has resulted in two sub-types of carbureters. If the float is in a separate and distance chamber, at one side of the jet, the carbureter is of the "eccentric float" type, but if the float chamber surrounds the jet and air passage, the carbureter is of the "concentric float" type. These two types are illustrated in Figs. 44 and 45 respectively.

Gasoline Adjustment. — The ratio of gasoline to air in a mixture that is drawn into a gasoline-engine must be readily changed to suit varying conditions due to atmospheric changes and also variations in the quality of the gasoline. This adjustment is usually made by

the needle valve in the spray nozzle mentioned above. Other methods have been used but the needle valve seems to be the most convenient and the most satisfactory.

Air Adjustment. — The air that passes around the jet of gasoline becomes saturated with the fuel, especially if this air be warm, as it is when taken from a jacket surrounding the exhaust pipe. Since this saturated air is too rich to form an explosive mixture, auxiliary air is drawn in to dilute the charge between the carbureter and the engine. The quantity of auxiliary air is regulated to suit the existing conditions, by means of a butterfly valve or an automatic lift valve. It is the difference in pressure between the fuel and the atmosphere which causes both the air and the fuel to flow. As the engine speeds up this pressure difference increases and hence the flow of both air and fuel increases. Unfortunately the flow of liquid fuel increases faster than that of the air, causing a richer mixture as the speed increases. For this reason the auxiliary air valve is required and it must allow more and more air to flow as speed increases.

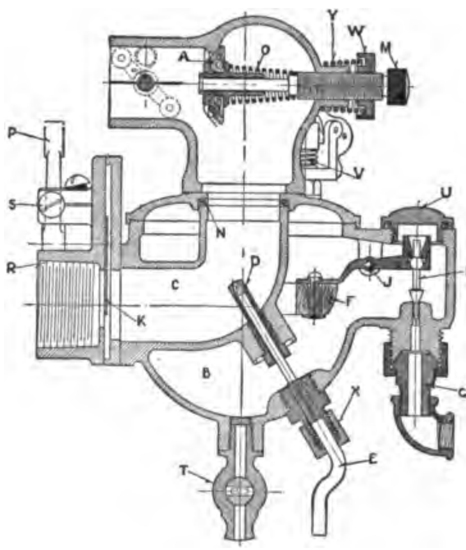


FIG. 46. — Schebler Carbureter.

Schebler Carbureter. — The carbureter shown in Fig. 46 is of the concentric float type, this float being of cork, shown at *F*. The gasoline enters through the needle valve *H*, which is controlled by the float, keeping a constant level in the float chamber. Gasoline enters the air pipe through the nozzle *D* which is slightly higher than the gasoline level. Air enters through and around valve *A*, passes down over the gasoline jet and out at *R*, whence it is piped to the cylinder. The throttle *K* is controlled by a rod attached at *P*. The air supply may be adjusted by the butterfly valve in the air inlet, and the gasoline by the lever *E* on the needle valve.

Renault Carbureter. — Fig. 47 illustrates the type of carbureter used on Renault motor cars. It has several unique features, particularly the arrangement for controlling the auxiliary air supply. Gasoline enters the chamber 2 and rises in the float chamber, the height being controlled by the copper float 4. Chamber 13 is also filled with the gasoline to the same level, through passage 26. Primary air is taken from a jacket on the exhaust

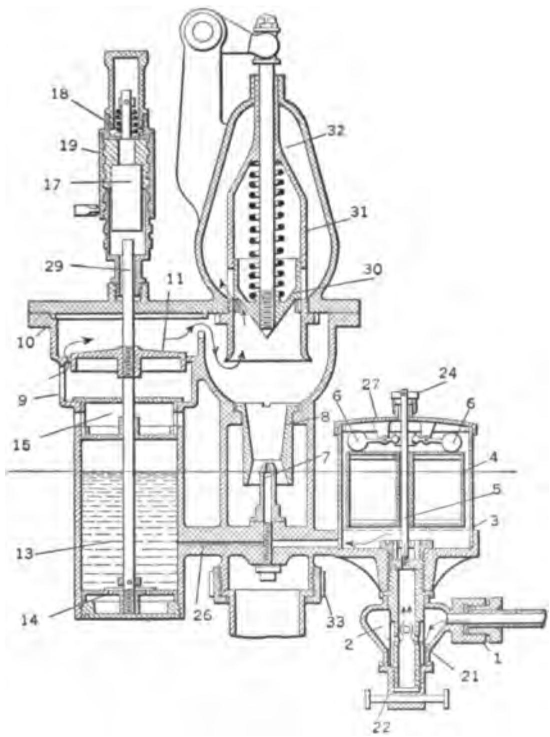


Fig. 47. — Renault Carbureter.

pipe, enters the carbureter at the bottom and passes up through the Venturi tube, vaporizing the gasoline at the throat. Auxiliary air is admitted through the valve 11, forming the base of a supplementary air chamber communicating freely with the mixing chamber above the Venturi tube. At the lower extremity of the stem of the air valve is a piston capable of an up-and-down movement in the cylindrical chamber containing gasoline. As the piston meets with a certain resistance by reason of the gaso-

line, rapid vibrations common to spring controlled valves are avoided.

About $\frac{1}{8}$ inch above the upper extremity of valve stem 29 is a counterweight 17. The initial opening of the valve is, therefore, made without any difficulty, the only resistance to overcome being that of the piston within the gasoline tube. As soon as the stem comes in contact with the balance weight, however, further opening can only be made by overcoming the weight of the balance and the spring 18 to which it is attached. The weight of the balance and the tension of the spring are calculated to give a correct lift and in consequence a correct amount of air for all engine speeds.

Throttling is done in the upper portion of the mixing chamber. The arrangement consists of a movable sleeve 30 on the lower portion of which are a number of elongated grooves corresponding with holes in the cylindrical vessel 31. The internal sleeve is mechanically operated by means of the foot accelerator, and in proportion as it is raised more of its grooves are brought into relation with the holes, allowing a larger passage of the mixture to the chamber 32, and from there into the intake pipe. Further throttling is obtained by a small lever on the steering wheel, by means of which it is possible to cut off the supply of gas entirely.

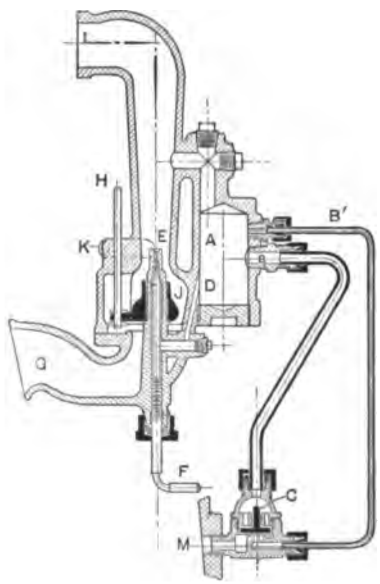


FIG. 48.—Mixer Used on Stationary Gasoline-engines. Seager Gas-engine Works.

For ease in starting a small dashboard lever is fitted by means of which the counterweight 17 can be lowered until it comes in contact with the stem of the supplementary air valve, thus shutting off the supply of auxiliary air and giving a very rich mixture. As soon as the motor has been started the lever is carried over to its original position, allowing the counterweight and spring to rise normally.

Mixer — Stationary Engine. — The mixer used on gasoline-engines built by the Seager Gas Engine Works of Lansing, Michigan, is shown in Fig. 48. The pipe L is connected to the inlet valve chamber, and air for combustion is drawn up from G. This produces a suction in chamber A which tends to draw up gasoline from the tank at M. The check valve C lifts during the suction stroke, closing pipe B but opening pipe B', thus allowing the gasoline to be drawn up into the chamber A. Pipe B acts as an overflow, down which the surplus gasoline may flow during any stroke of the engine except the suction.

Gasoline flows from chamber A to the nozzle E, which is about $\frac{1}{8}$ inch higher than the overflow. This spray nozzle is adjusted by the needle valve F. In order to enrich the mixture for starting, the weight J is lifted by the rod H, closing off part of the air in the Venturi throat. The spring K holds H with enough force so that J will not return by gravity.

CHAPTER XI

GOVERNING — THEORY OF WATT GOVERNOR — SYSTEMS OF GOVERNING — TYPES OF GOVERNORS

Governing. — Governing is the process of changing the power generated in the cylinder to correspond to the changes in load. The governor, the apparatus that effects this change in power, should be beyond criticism as to reliability, for on the governor depends the success of the engine. The governor must be sensitive to speed changes and must vary the power immediately to suit the speed changes. The operation of the governor must be smooth and without fluctuation.

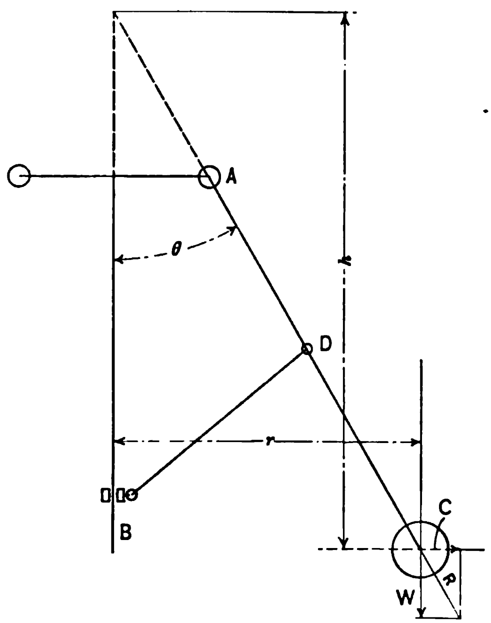


FIG. 49. — Diagram of Watt Governor.

The governor proper, on a large percentage of engines, consists of two balls or weights, one on either side of a spindle, which revolve at a high speed. These balls or weights are connected by a system of links to a sliding collar on the spindle. As the spindle and balls revolve, the centrifugal force of the balls projects them in a radial direction, and by this movement the sliding collar is lifted. To this collar is also attached the mechanism for changing the power in the cylinder.

This simple governor, called the Watt governor, is illustrated in Fig. 49, which shows only one of the balls. For purposes of calculation it is generally assumed that the weight of the two balls

is concentrated in one, which assumption does not affect the results of the calculation. The balls rotate about the spindle with an angular velocity proportional to the speed of the engine. If the engine speed increases the balls move outward about A and the collar B is lifted. Since the link AD can turn freely about A, then, neglecting friction, it is necessary for equilibrium that the resultant of the weight W and the centrifugal force of the ball (that of the arm being neglected), should act in the direction of the link. If we take W vertical, then draw a horizontal line to meet the direction of the link and complete the parallelogram, C is the centrifugal force for that position of the governor. If the governor is revolving at N revolutions per minute, and ω is the angular velocity in radians per second,

$$\omega = \frac{2\pi N}{60},$$

$$C = \frac{W}{g}\omega^2 r = \frac{W}{g}\frac{4\pi^2 N^2 r}{60^2} = \frac{WN^2 r}{2940},$$

$$\frac{h}{r} = \frac{W}{C},$$

$$h = \frac{2940}{N^2}\text{ feet} = \frac{35{,}280}{N^2}\text{ inches},$$

$$N = \frac{54.2}{\sqrt{h}}\ (h\text{ in feet}) = \frac{187.8}{\sqrt{h}}\ (h\text{ in inches}).$$

If h_2 and h_1 are the heights to which the governor is limited by the stops which define the travel of the sleeve, then

$$N_2 - N_1 = 54.2\left(\frac{1}{\sqrt{h_2}} - \frac{1}{\sqrt{h_1}}\right)$$

is the range in speed within which the governor can exercise control. The mean speed is $N = \frac{N_2 + N_1}{2}$ and the variation from mean speed within the range is $\frac{\frac{1}{2}(N_2 - N_1)}{N}$ or $\frac{N_2 - N_1}{N_2 + N_1}$ expressed as a fraction of the mean speed or $100\,\frac{(N_2 - N_1)}{(N_2 + N_1)}$ as a percentage of the mean speed.

For ordinary cases the movement of the balls of a Watt governor corresponding to a given variation in speed is small, and the governor is comparatively insensitive. Thus, if the normal speed

of the governor is 100 and a variation of 2 per cent from the mean is allowed, so that the maximum and minimum speeds are 102 and 98, the corresponding heights will be

Revolutions per minute =	102		100		98
h =	3.39		3.53		3.67 inches
Differences		0.14		0.14	

The Watt governor is stable in the sense that there is a definite position corresponding to any speed within its range. If the speed increases, C increases and the balls move outwards a definite distance, satisfying the condition that $\tan\theta = \frac{C}{W}\cdot$ In order that the governor may completely control the gas supply, the mechanism between the sleeve and the gas-valve must be so arranged that, as the sleeve moves from the lower to the upper stop, the valve will move from full open to shut.

Systems of Governing. — The governor may regulate the speed in various ways. One of the most familiar for small engines is the **hit-and-miss** system, in which explosions are discontinued when the load lightens and the engine speeds up. The second, in importance, is the **throttling** system in which the mixture is throttled between the carbureter or mixing valve and the inlet-valve. A third system is the **constant quantity** in which the proportion of gas and air are changed by the governor, and there is no throttling effect. Another system, which is very like the throttling, is the **constant quality,** in which the proportion of gas and air are unchanged, but the quantity is varied to suit the load. This result may be obtained by other methods than throttling.

Hit-and-miss Governing. — It has been stated elsewhere in this book that gas-engines are not efficient at light loads. The reason is this: the cylinder volume is designed to give a certain maximum power and to generate this power the cylinder must be filled with gas and air in the proper ratio to secure best results. When this cylinderful of mixture is compressed into the clearance space, the pressure attained should be the proper pressure for obtaining complete combustion. If any other conditions exist, that is, if less than a cylinderful of mixture is drawn in and compressed, the power generated will be less but efficiency will be lowered, as the best conditions, those for which the engine was designed, do not prevail.

If a single-cylinder, single-acting, four-cycle engine with hit-and-miss governing be designed for 230 revolutions per minute, the maximum number of explosions would be 115 per minute. Allowing 15 per cent overload, the number of explosions at the rated load would be 100, at $\frac{3}{4}$ load 75, $\frac{1}{2}$ load 50, etc. All the explosions occur when a full charge is in the cylinder, hence combustion takes place under the best conditions, and efficiency is high, even at light loads.

There are several methods by which an engine can be made to miss an explosion; some are good and others are not. The simplest method is to break the ignition circuit for the required number of strokes, but this is not a good way as the very important advantage of the hit-and-miss governing, that of high efficiency at light loads, is lost. Even if the spark is cut out fuel is drawn into the cylinder discharged into the exhaust unburned. This is a source of danger as this unburned fuel often will explode when brought into contact with the hot muffler or exhaust pipe.

Another method, the one generally in use, is to allow the inlet-valve to remain closed during the suction stroke. If this valve be opened mechanically, the usual method is to have the cam-roller connected to the governor by a system of links. As the speed goes above the normal, the governor moves the roller out of line with the cam and the valve remains closed for that stroke or until the speed of the engine drops sufficiently to allow the roller and cam to engage again.

When the above method is used, some means should be provided to allow air to be drawn into the cylinder during the suction stroke. In one make of engine two inlet-valves are used, one for air and one for gas, but both are opened by the same cam. The spring on the air-valve is light enough, however, so that when the governor throws the roller free from the cam, the air-valve opens, due to the partial vacuum in the cylinder, while the gas-valve remains closed, owing to the stiffer spring on that valve.

Where an automatic valve is used, that is, one that opens due to the decreased pressure in the cylinder during the suction stroke, it is not possible to use the apparatus just described. Instead, there is an auxiliary cam next to the exhaust-valve cam, which, in normal operation, is not in use. When the speed increases above normal, the governor moves the exhaust-roller so that it engages this auxiliary cam, which then opens the exhaust-valve during the

suction stroke as well as the exhaust stroke. Since the exhaust-valve is held open, there is no vacuum produced in the cylinder during the suction stroke, no charge enters and an explosion is missed.

One disadvantage of this type of governor mechanism is that long exhaust pipes may produce a partial vacuum in the cylinder due to the inertia of the column of gases rushing out of the pipe. The vacuum thus produced may be sufficient to open the inlet-valve slightly so that a small amount of fuel is let in and discharged into the exhaust without burning. Naturally, the fuel thus lost helps to reduce the efficiency of the engine.

Hit-and-miss Mechanisms. — Mechanisms used in connection with hit-and-miss operation are plentiful in number and vary widely in construction. Those using a fly-ball governor to move the parts are very simple. There is usually a trigger, called the

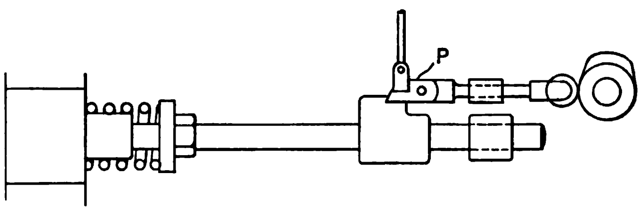


Fig. 50. — Hit-and-miss Mechanism Used in Connection with Watt Governor.

pick-blade, which is under the control of the governor as regards position. This pick-blade is moved by the cam and taps the valve rod if the valve is to be opened. If the valve is to remain closed, the governor moves the pick-blade out of the line of action. Fig. 50 illustrates this type.

Fig. 51 illustrates the hit-and-miss governor mechanism used on the smaller engines built by the Seager Gas Engine Works. The cam C operates the exhaust-valve stem D through the bell crank. Attached to the bell crank is the arm carrying the notch N, which oscillates with the bell crank, moving to the right as the exhaust-valve opens. The trigger A and pick-blade B form a bell crank, pivoted near E. As the governor weight W revolves with the flywheel, the finger F forces A to the left and hence lifts B, so that when the speed is normal, B cannot engage in N. However, when the speed increases, W moves radially and F

moves towards the center of the shaft, allowing A to go to the right and B to drop into the notch N, **when the exhaust-valve is open.** This keeps the exhaust-valve open during the succeeding suction stroke so that no charge is drawn in. B remains in N, keeping the exhaust-valve open, until the speed lowers and F forces A to the left once more. Even then B cannot escape from N until the exhaust cam gets into the open position and relieves B of the tension of spring S_3.

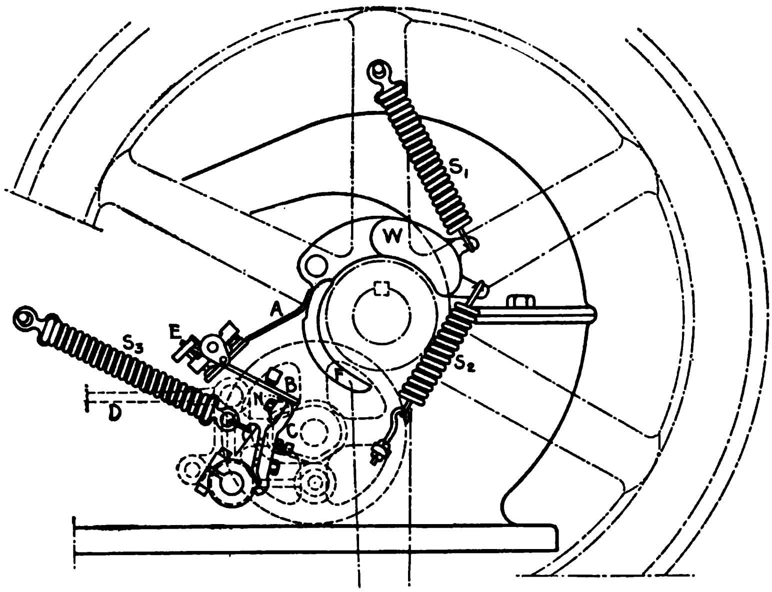


FIG. 51.—Centrifugal Mechanism for Hit-and-miss Governor. Seager Gas Engine Works.

Fig. 52 illustrates a type of inertia governor built by Crossly Bros., Ltd., Manchester, England, and used on their small four-cycle engines. The cam A actuates the roller B at the proper time for opening the inlet-valve. The governor arm G carries on one end the weight E and on the other the pick-blade H. The valve-rocker D, which transmits the motion from the roller B to the valve-stem L, is fulcrumed at C, and carries the governor arm hinged to it at F. The governor arm, with the weight and pick-blade, is thus free to swing through a small arc relatively to the rocker D, except that the spring S_3 holds it against the rocker and

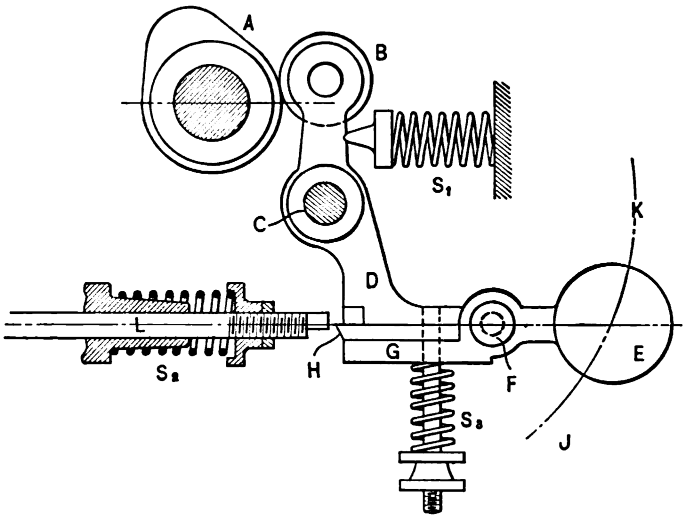


FIG. 52.—Inertia Type of Hit-and-miss Governor.

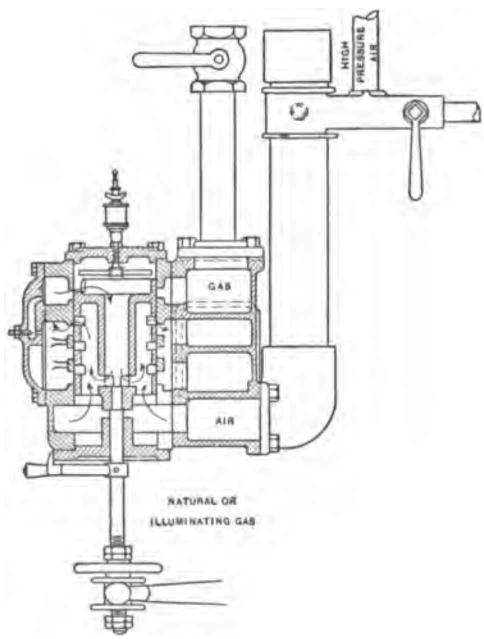


FIG. 53.—Mixing and Throttling Valve. Bruce-Macbeth Engine Co.

permits the pick-blade, under normal conditions, to meet the end of the valve stem L. S_2 is the main valve spring and S_1 holds the roller B to the cam.

Under normal operation the weight E moves in the arc JK. However, when the speed increases, the inertia of the weight E causes it to lag behind so that it revolves about the center F. H then does not come in contact with the end of L, and the inlet-valve remains closed for that stroke.

Throttling Governor. — In Fig. 53 is illustrated the mixing and throttling valve used by the Bruce-Macbeth Engine Company of Cleveland, Ohio. Gas enters from the pipe marked "gas" through the port in front, as shown by the arrow, and passes down through the center of the piston-valve. At the bottom of this valve it mixes with the incoming air at practically atmospheric pressure. The mixture then passes up the annular space in the valve and out through the three ports to the inlet-valve chamber between the passages marked "gas" and "air" respectively.

The piston-valve has no rings but is ground to a fit. The only adjustment is at the handwheel shown, which changes the proportion of gas to air. The three ports shown do not open and close together; in fact, the top port is cut on a $\frac{3}{16}$-inch spiral. At a point where this spiral is fully opened, the second port begins to open and after this second port is open $\frac{1}{32}$ inch, a third port begins to open.

The needle-point screw in the cover of the valve chest allows gas to be by-passed from the gas-pipe directly into the inlet-valve chamber, around the throttle-valve. This hole is but $\frac{3}{16}$ inch in diameter for a 150-horse-power engine, but a slight adjustment of the screw gives the proper mixture and consequently maximum efficiency.

Variable Lift. — Throttling is sometimes accomplished by means of a variable lift on the inlet-valve. This is usually done by using a movable fulcrum in connection with a valve rocker arm. This arrangement is illustrated in the Deutz valve gear, Fig. 54. A shows the position at full load and B at light load. The gas passage is above the air passage, and is closed by a separate valve on the main stem. The main valve controls the air admission. As the valve lifts, a rush of air enters the cylinder followed by a mixture of gas and air. If there is a slight play between the shoulder of the gas-valve and the sleeve above it, the

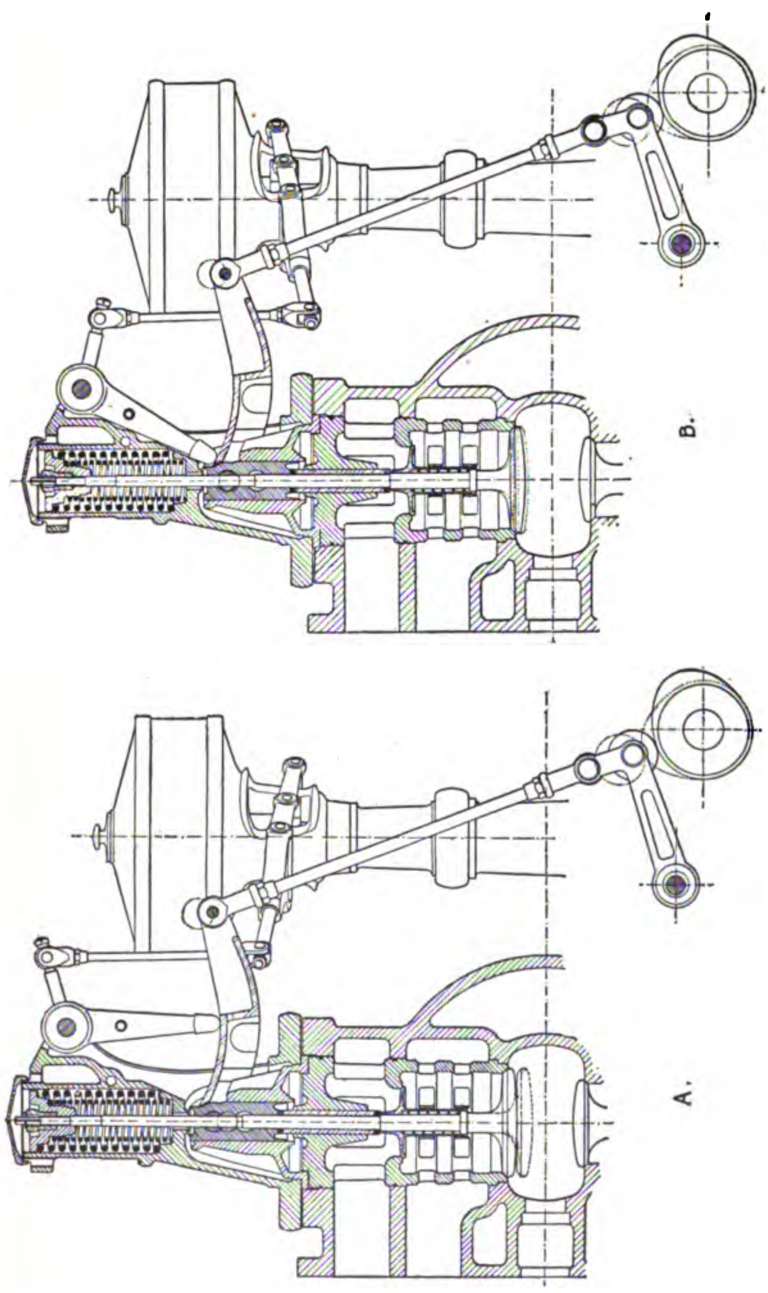


FIG. 54.—Deutz Valve Gear and Governor.

main valve will open slightly ahead of the gas-valve but the gas-valve will close before the main valve. It will be noticed that there is a small spring within the gas-valve sleeve which keeps

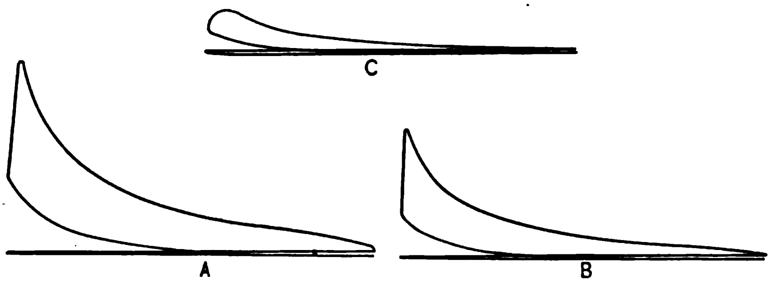


FIG. 55.—Indicator Cards from Engine Equipped with Deutz Valve Gear and Governor.

this valve against the seat until forced off by the sleeve on the main stem.

This gear is practically a constant-quality device except at very

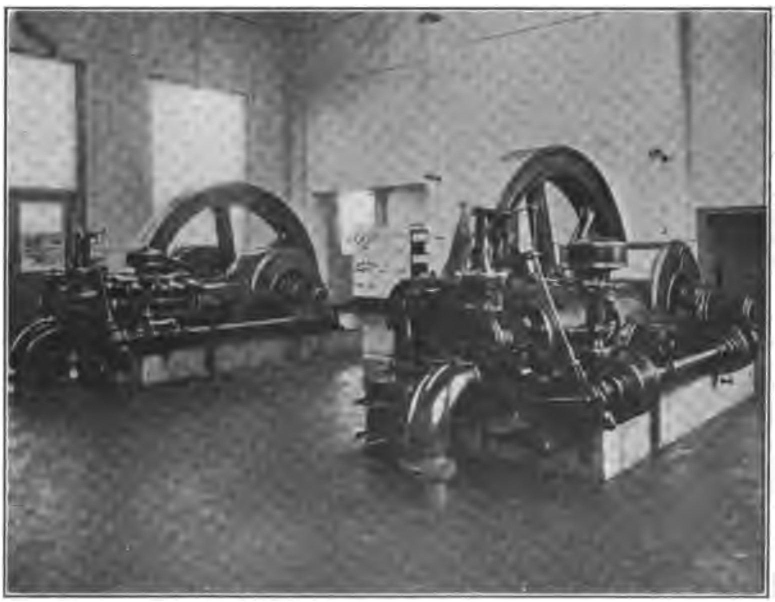


FIG. 56.—Deutz Gas-engines, Wetzlar, Germany.

light loads. The variation in quantity, and hence in the compression, is clearly shown in Fig. 55, which illustrates cards taken from a Deutz engine at full, half and light loads. The appearance

of this gear on the actual engine is illustrated in Fig. 56, which shows two 100-horse-power Deutz engines in the power house of the railway station at Wetzlar, Germany. These units operate on producer gas made from the refuse taken from locomotive smoke-boxes.

Another form of constant-quality governor is illustrated in Fig. 57. This shows a combined mixing- and inlet-valve *A* and *B* controlled by the eccentric *G* through rolling levers. The mixture

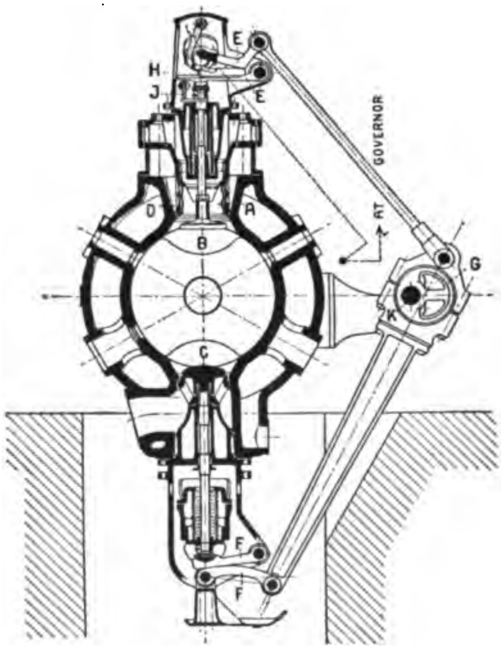


Fig. 57.—Valve Gear and Governor. Machinen-Fabrik Augsberg-Nürnberg A. G.

valve *A* regulates the gas and at the same time the air, by means of an air slide *D* rigidly connected to the gas-valve. The inlet-valve *B* is of the ordinary mushroom type, and regulates the quantity of mixture admitted to the cylinder.

The governor alters the lift of the combined mixture and inlet-valve by moving a die *H* which is inserted between the upper and lower inlet-valve rolling lever *E*, and in this manner the quantity of mixture drawn in is suitably proportioned to the required output. The composition of the mixture is practically

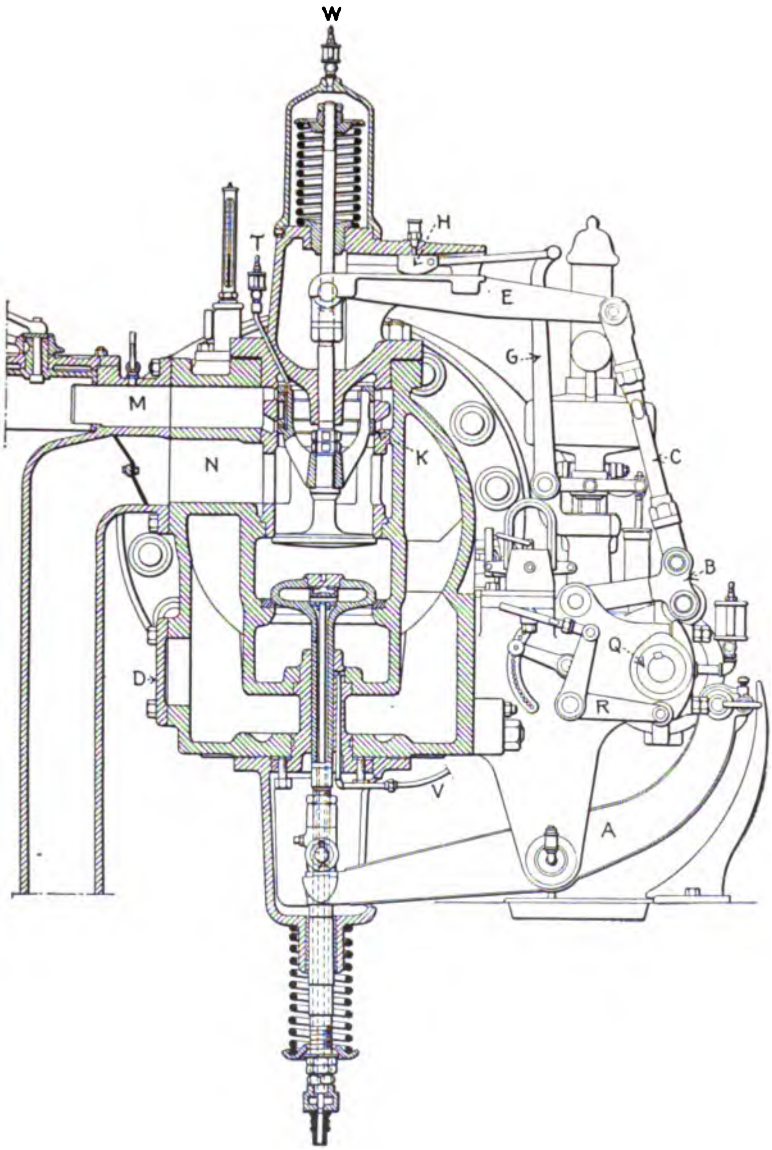


Fig. 58.—Valve and Governor Mechanism. Seager Gas Engine Works.

the same on all loads after the valves have been set for the most efficient combustion, according to the kind of gas used. If the gas changes, then the ratio of air to gas is altered by hand while the engine is running, by turning the air slide valve *D* by means of a lever.

With gas blowing-engines for blast-furnaces, the dies between the inlet rolling levers are not moved by the governor, but by a handwheel. Racing of the engine is then prevented by a small safety governor which switches off the ignition current.

The valve and governor mechanism of the Olds engine, now built by the Seager Engine Works, is illustrated in Fig. 58. In this engine there is a combined gas- and inlet-valve on the same stem, the inlet-valve being of the ordinary mushroom type, while the gas-valve is the sleeve *K* above the inlet-valve. This gas-valve has a lap so that the main valve opens appreciably before the gas is admitted. The fulcrum block *H* is moved out and in by the governor. At low speed full load the block is to the right, giving maximum opening to the valves. As the speed increases the block is moved to the left and the lift of the valve is decreased. As the lift decreases, the ratio of gas to air becomes smaller, or the mixture becomes leaner. The quantity decreases, also, so that at light loads the compression is much lower than at larger loads.

Constant-quantity Governor. — In constant-quality governing, whether it is done by actual throttling, by cut-off, or by changing the lift of the valve, compression at light loads decreases very appreciably, enough so that the ignition is uncertain, and often fails entirely for several strokes in succession. The author has made tests on engines which apparently indicated higher power at light loads than at heavier loads. The reason for this was that on light loads the ignition would fail entirely for a stroke or two and the engine would slow down enough to get a charge of good gas, then an explosion would occur that would give a higher mean effective pressure than the average. In order to get the proper indicated power it was necessary to count the missed explosions as on a hit-and-miss engine.

In order to overcome this difficulty of missing explosions at light loads, several forms of governors have been designed. One of these is the one used on the Buckeye engine and is illustrated in Fig. 59. The exhaust-valve at the bottom and the inlet-valve at

the top of the cylinder are operated by the same eccentric through rolling levers. Each combustion chamber is supplied with fuel by a separate mixing chamber. The gas enters the lower compartment of each mixing chamber through the small pipe enclosed

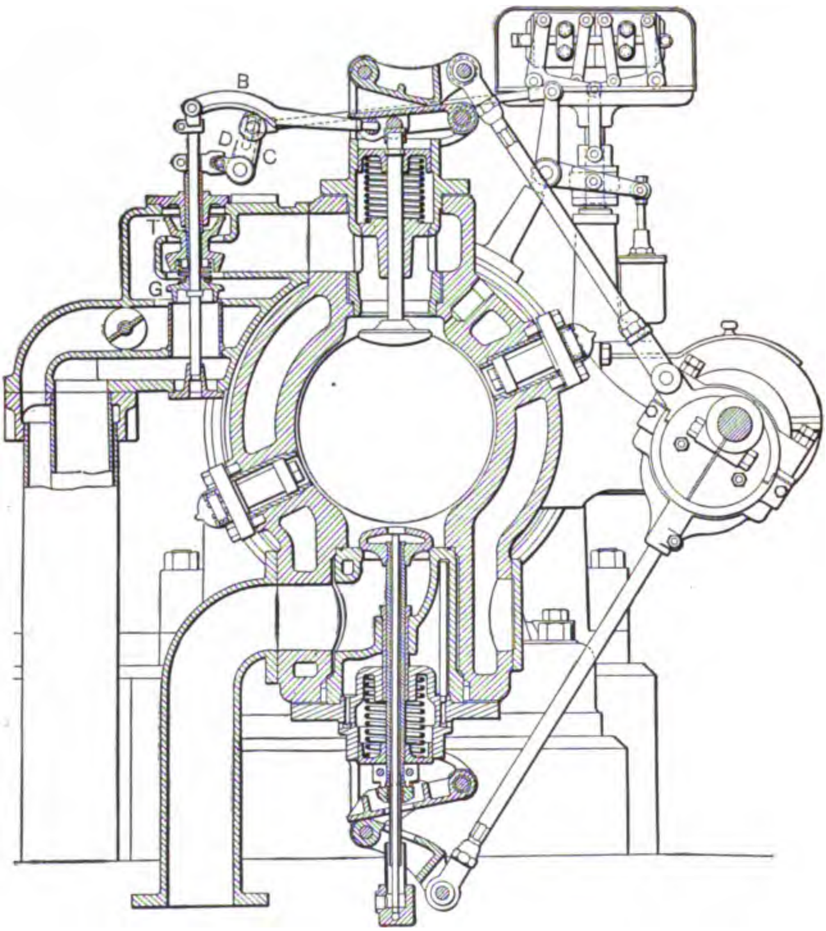


FIG. 59.—Buckeye Valve and Governor Mechanism.

in the air pipe. A gas-valve *G* prevents the gas and air from mixing at all times except during the suction stroke. Valve *G* derives its motion from the main inlet-valve lifter through the segment lever *B* which fulcrums on a roller carried by lever *C*. This lever is under the control of the governor with the result that

while the gas-valve lifts every suction stroke, the extent of the lift is variable, being greater at heavy loads and less on light loads. Surrounding the gas-valve stem is a tubular stem attached to the balanced throttle-valve *T*. This valve is also governor-controlled through the lever *D*. It should be noted, however, that while the gas-valve opens and closes every suction stroke, the lift varying with the load, the throttle-valve is held stationary as long as the load is constant and moves to a new position only when the load varies. At all loads above 60 per cent of the rating, this valve is held so high above its seat that the mixture is not throttled appreciably, the governing being done entirely by varying the lift of the gas-valve. As the load falls off, the increased speed causes the governor weights to move the fulcrum roller under the segment lever to a position where the gas-valve will have a decreased lift; and at the same time the action of the governor reduces the opening of the throttle-valve. All loads from maximum down to about 60 per cent are handled purely by varying the strength of the mixture, while the total volume of the mixture and, consequently, the amount of compression remain unchanged during this part of the load range. At about two-thirds load the throttle-valve begins to reduce the volume of mixture admitted to the cylinder so that all lighter loads are carried by weakening the charge and using less of it.

Since lean fuel mixtures burn more slowly than rich ones it has been found advisable to vary the ignition timing as the load on the engine varies. The governor does this automatically by advancing the point of ignition on light loads and retarding it on heavy ones.

Advantages of Different Governing Systems. — For small engines driving machinery which does not require particularly close regulation, the **hit-and-miss** system is to be preferred. It is cheap and simple to build, and, being simple, does not get out of order easily. On a four-cycle single-cylinder engine one missed explosion means that the engine must make eight strokes or four revolutions with one power stroke, which shows conclusively that close regulation cannot be expected.

The plain throttling governor is next to the hit-and-miss as regards simplicity and low cost. If the throttle-valve admits the proper amount of correct mixture at full load to give maximum efficiency, at loads less than the full load, the amount ad-

mitted will be less and efficiency will drop off rapidly. Fig. 60 shows two cards from a throttle-governed engine, one at full and one at a lighter load. The one at full load has a pressure at the beginning of compression of 14 pounds, P'', and a pressure at the end of compression of 125 pounds, P'. The light-load card has pressures at these points of 10 and 95 pounds, respectively. The ratio of full mixture, by weight, at beginning of compression is 1.4. The mean effective pressures of the two cards are 60 and 20 respectively, or the large card shows 3.0 times the work of the small card on 1.4 times the fuel. Hence the efficiency of the large

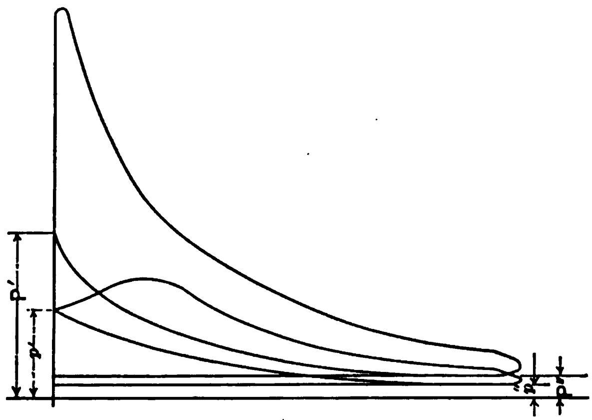


FIG. 60.

card is $\frac{1}{1.4} \times 3.0 = 2.14$ times as great on the full load as on the light load.

The loss in efficiency here is due to filling the cylinder only to 10 pounds absolute pressure at the beginning of compression instead of to 14 pounds as on the full load. It will be noticed that a difference of 4 pounds at the beginning of compression makes a difference of 30 pounds at the end of compression. One way to avoid part of this loss is to dilute the mixture at light loads with enough air to increase the compression up to the normal.

Fig. 61 illustrates two cards showing the effect of diluting the mixture at light loads to bring up the compression pressure. Little is gained here as the increased dilution practically offsets the advantage gained by increased compression, and slow burning is the result as before.

The method employed in the governor shown in Fig. 59 is to advance the spark and start combustion earlier at light loads. Since the weaker mixture is slower burning than the normal mixture, early ignition is the remedy for the late burning shown

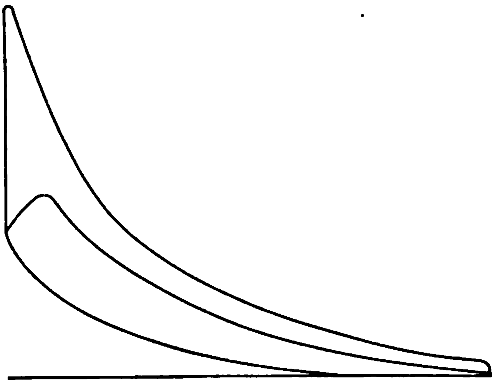


FIG. 61.

in the cards in Figs. 60 and 61. The effect of advancing the spark and making ignition earlier is shown in Fig. 62.

Still another method which has been tried, but has not been found entirely satisfactory, has been to use more of the mixture at

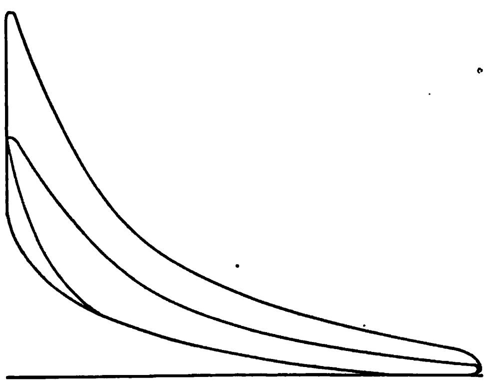


FIG. 62.

light loads than at full load, to insure higher compression at the light loads. The only way to do this is to cut down the amount of mixture at full loads by throttling, thus reducing the pressure at which compression begins. If, by this method, proper com-

pression and therefore maximum efficiency is secured at light loads, the compression at full loads will be below that required to give best efficiency. Thus, while efficiency is increased at one point, at a more important point it is decreased.

Power Governors. — In large gas-engines, particularly the twin-tandem type, the power of the centrifugal governor is insufficient to operate the valves and other means must be resorted

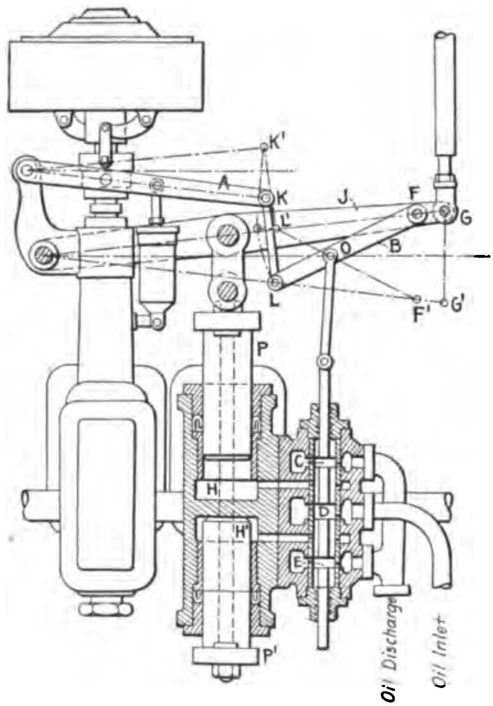


FIG. 63. — Power Governor Used on Large Gas-engines.

to. In such cases the governor proper usually operates a small pilot valve which in turn admits oil under pressure to plungers which move the governing valves. In Fig. 63 the governor operates the pilot valve *CDE*. The plungers *P*, *P'*, actuate the lever *J*, which regulates the engine valve gear. The governor is shown in its low position, the governor lever *A* is down and the hydraulic plungers are held up by the pressure of oil which has been admitted above *D* and into *H*. If speed increases and the governor

rises, it will lift the pilot valve. The oil under pressure will then be admitted below D and into H'. At the same time the oil in H will be discharged through the upper port which is uncovered by spool C. The hydraulic plungers will thus descend, bringing down the lever J, and the pin F moves towards the position F', until the pilot valve again covers the three ports and stops all motion of the governor. This position is held until another change of speed occurs, when the pilot valve will be moved up or down as the speed increases or decreases, the plungers being moved in a corresponding direction. Thus at any given speed, the governor has a fixed position.

CHAPTER XII

COOLING ENGINES

Necessity for Cooling — Cooling Systems

General. — It is necessary to furnish some means of cooling at least the cylinders of all internal combustion engines. The amount of heat generated in the cylinder of such an engine varies from 7500 to 15,000 B.t.u. for each horse-power hour developed. From 25 to 35 per cent of this heat must be conducted away and, within certain limits, the lower the temperature at which conduction takes place the better will be the conditions for the engine.

The amount of heat that will flow per unit of surface from a gas-engine cylinder depends on first, the time; second, the difference in temperature; third, the material of cylinder wall; fourth, the thickness of cylinder wall; fifth, the condition of cylinder wall on the inside and outside. We may select any unit of time for the discussion and since the hour is the most convenient, there need nothing more be said in regard to the time. To illustrate the other items, we may refer to the sketch in Fig. 64. The heat must first flow through the surface A into the wall of the cylinder B. This cylinder wall is usually of cast-iron and of a thickness varying with the diameter of the cylinder and the pressure. The condition of the cylinder wall inside is fixed by the necessity of lubrication, so that the only condition which is really variable is the outside surface of the wall. If the wall C is left smooth, or, in other words, if C and A are of the same area, and the air surrounding C is still, the heat that will flow per hour from A to B and from B to C would not take care of the excess that must be radiated, unless the temperature difference between C and the air is very great, so great in fact that the metal becomes injured.

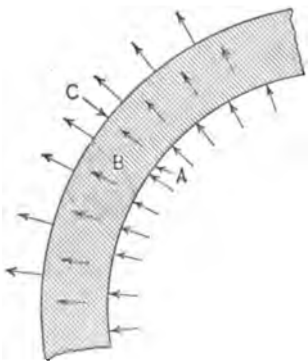


FIG. 64.

If we change the character of the surface C by roughing it up and increasing its area, keeping the area of surface A constant, we have a condition shown in Fig. 65. With this condition more heat would flow from surface C than from surface A, since for one square foot of A we might have several square feet of C. But since the heat is constant, that is, since as much flows from C as into A, another condition must change. This condition is the temperature of the surface C. Since there is more outlet, as it were, for the heat at C, the temperature of B will drop lower in Fig. 65 than in Fig. 64. To further increase the flow of heat from C to the air, the air surrounding C is caused to move. This is usually done by means of a fan.

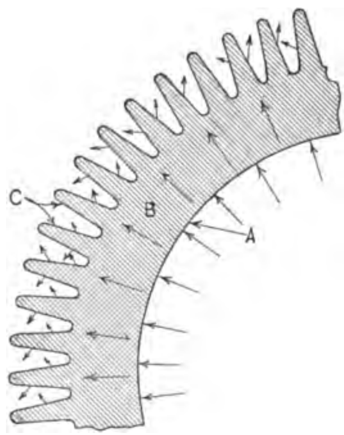


FIG. 65.

Air-cooled Engines.— Small engines may be cooled successfully by increasing the area of the outside surface of the cylinder as illustrated in Fig. 65. The area is increased in many ways. One of the first air-cooled automobile engines built in the United States was made with numerous copper spines, the size of a lead pencil, screwed into the cylinder wall, until the cylinder presented a grotesque appearance, not unlike a mechanical porcupine. Another method is used to cast deep, thin ribs on the cylinders. In some cases these ribs extend around the cylinder transversely as in the motor cycle engine, while in others the ribs are longitudinal, parallel to the axis of the cylinder.

In Fig. 66 is shown an air-cooled engine, built by the "New Way" Motor Company of Lansing, Michigan. The cylinders have cooling fins arranged so that they are perpendicular to the axis of the cylinder. These fins are perforated with holes from $\frac{1}{4}$ to $\frac{3}{8}$ inch in diameter. The fins are also connected by bars or rods running at right angles to them. Cooling fans force air through these fins, and the bars and the holes break up the air currents so that the air is heated uniformly and no stream lines can exist. The cylinder is surrounded by a sheet-metal jacket which fits closely against the fins, so that the air that is forced

in by the fan must pass between the fins and through the holes.

Experiments have shown that the best results are obtained on this engine when a thermometer inserted between two fins on the cylinder head registers between 300 and 350° F. When one of the engines was operated on a 14-day full-load test, the temperature did not drop below 310 degrees, nor rise above 340 degrees, as shown by half-hourly readings.

The Franklin Automobile motor shown in Fig. 88 is cooled by air. Longitudinal fins made of sheet steel are cast into the cylinder walls. These fins are encased by a jacket of steel, open at the top and bottom. The rim of the flywheel of the engine carries

FIG. 66. — "New Way" Air-cooled Engine.

fan blades of the Sirocco type. The blades draw air down between the fins and their surrounding jackets. The hood that covers the engine is arranged in such a way that there is no path left for the air except through the path inside the jackets.

Water Cooling. — Some manufacturers of small engines, and all manufacturers of large engines, cool the engine cylinders with water. When water is used for cooling, the cylinder is made with a double wall, the space between the two walls being the **water-jacket.**

Single-acting engines are usually cooled only around the barrel of the cylinder and on the cylinder head. When the diameter of the cylinder is above 20 inches, however, the piston must be

cooled. In double-acting engines, both the piston and rods must be water cooled, as there is no chance for radiation as there is in a single-acting engine.

Water cooling is more effective than air cooling for this reason: The heat conducted away from a surface surrounded by water amounts to about 100 times more than that conducted away from

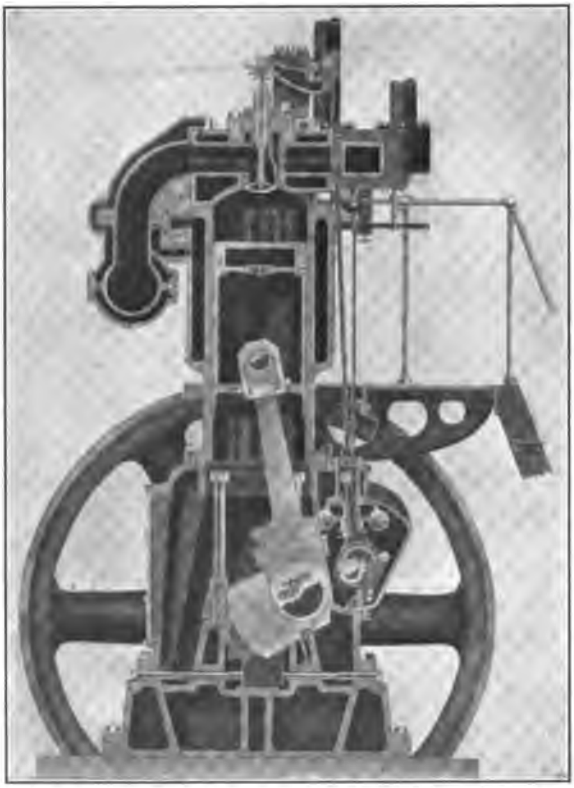


Fig. 67. — Water-cooled Vertical Gas-engine.

a similar surface surrounded by air. Of course this ratio depends entirely on the motion of air and water. Much more heat is conducted away by air in motion than by still air.

Fig. 67 shows plainly how the water-jackets are arranged in a vertical engine. It will be noticed that the inner and outer cylinder walls are cast in one piece, and that the jacket extends down about to the location of the top of the piston in its lowest position.

It is also evident that the cylinder head and the exhaust manifold are water jacketed.

In larger engines the outside wall of the cylinder is cast separate from the inside wall. This is done to prevent shrinkage strains in the castings and complicated core work. Fig. 68 shows one method of inserting this **liner,** as it is called. The liner is bolted rigidly to the outside wall at the upper end. A copper gasket inserted at *A* prevents leakage at that point. At the lower end of the cylinder a stuffing-box may be formed. This makes a water-tight joint and allows for expansion of the liner.

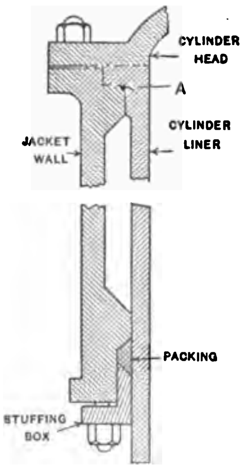


FIG. 68.

Pistons and Rods. — As stated above the pistons and piston-rods of double-acting engines must be water cooled. To accomplish this the rods and pistons are made hollow as shown in Fig. 69, which illustrates the arrangement of rods, etc., in the tandem double-acting engines built by the C. & G. Cooper Company of Mt. Vernon, Ohio. Water enters at the intermediate crosshead and flows in both directions, that is, toward the main crosshead, through the front piston, and toward the rear crosshead, through the back piston. The water enters the pistons at the bottom and on becoming heated rises and leaves by the top. At the front and rear crossheads the water escapes into long, trough-shaped funnels.

The water inlet at the intermediate crosshead gives trouble if not constructed with care. The best type appears to be a telescopic pipe, the inner or moving pipe being of bronze. A generous stuffing-box must be provided and air chambers used to prevent surges of water due to the pumping action. The water pressure for such an engine should be at least 40 pounds, and where water at that pressure is not available, a triplex pump, driven from the main engine, may be used to drive the water through the cooling system.

Cooling the Water. — In stationary engines no attempt is made to cool the water and use it over again except where the cost of the water makes the installation of a cooling system a neces-

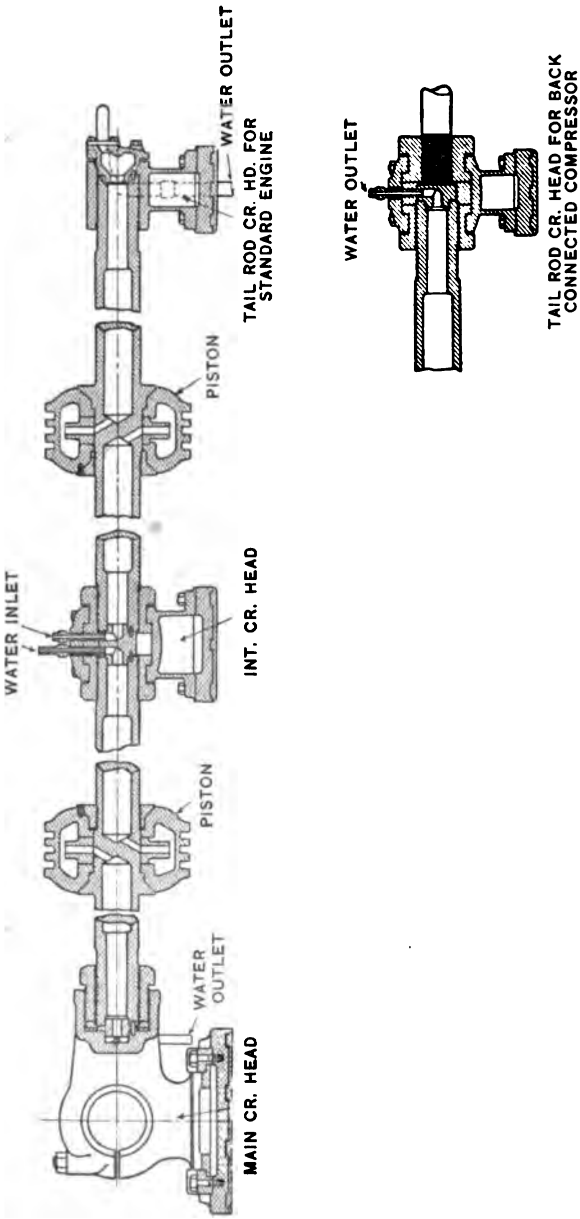


Fig. 69.—Water-cooled Pistons and Rods. C. and G. Cooper Co.

sity. For large engines cooling towers, similar to those used in connection with condensers, may be and are used. Some makers of small engines carry out the same idea by using conical towers made of galvanized iron. For automobile work the well-known cellular or tubular radiator is used. These are also sometimes used with portable engines as shown in Fig. 70, which illustrates a portable three-cylinder, vertical air compressor driven by a four-cylinder vertical gasoline-engine. The large rectangular radiator is shown at the front of the machine.

Water Required. — A very simple calculation will suffice to determine the amount of water required to cool a gas-engine. For

FIG. 70. — Portable Gasoline-engine Driving Air Compressor, Jacket Water Cooled in Radiator.

an extreme case let us assume 15,000 B.t.u. generated in the cylinder per horse-power hour, and that 35 per cent of this heat is absorbed by the jacket water. The amount of heat to be absorbed per horse-power hour will then be

$$0.35 \times 15{,}000 = 5250 \text{ B.t.u.}$$

We may assume that the water enters the jacket at 70° F. and leaves at 190 degrees. The rise in temperature will then be 120 degrees. Assuming the specific heat of the water to be 1, each pound of water will absorb 120 B.t.u. The water per horse-power hour will then be

$$\tfrac{5250}{120} = 43.7 \text{ pounds.}$$

Recapitulating, we have

$$W = \frac{xH}{100\,(t_1 - t_2)},$$

where

W = weight of water per horse-power hour in pounds,
H = B.t.u. supplied to engine per hour, per horse-power,
x = percentage of heat absorbed by jacket,
t_1 = temperature of water leaving jacket, degrees Fahrenheit,
t_2 = temperature of water entering jacket, degrees Fahrenheit.

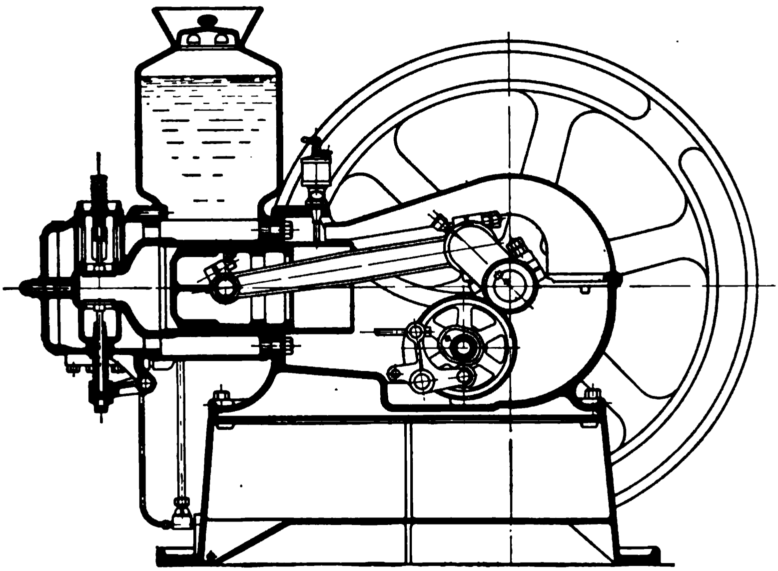


FIG. 71.—Hopper-cooled Engine. Seager Gas Engine Works.

If the water is not allowed to leave the jacket of the engine but is held there until it **evaporates,** much less water is needed as each pound of water or steam carries away with it the latent heat of vaporization in addition to the sensible heat. The formula for the amount of water necessary per horse-power hour becomes

$$W = \frac{xH}{100\,[(t_1 - t_2) + r]},$$

where r is the heat of vaporization at atmospheric pressure, approximately 970. With the conditions given in the previous problem, that is, with 15,000 B.t.u. generated in the cylinder per

horse-power hour, 35 per cent of which is absorbed by the jacket water, the water used per horse-power hour will be

$$W = \frac{xH}{100\,[(t_1 - t_2) + r]} = \frac{35 \times 15{,}000}{100\,(120 + 970)} = 4.8 \text{ pounds.}$$

An engine of the hopper-cooled type in which the water is evaporated is shown in Fig. 71. This is a 5-horse-power, four-cycle engine built by the Seager Gas Engine Works of Lansing, Michigan. The water consumption of this engine, based on above conditions, would be in the neighborhood of 24 pounds per hour.

CHAPTER XIII

POWER, EFFICIENCY, SPEED AND SIZE OF ENGINES

Brake and Indicated Horse-power

Power. — One definition of power is the " ability to do work." Thus, an engine that can do 33,000 foot-pounds of work in one minute is said to be one horse-power. Obviously, the same engine can do 550 foot-pounds of work per second.

Brake Horse-power. — If the power of any engine is expressed in terms of the horse-power actually delivered, that is, power that may be used to do actual work, it is called the **brake horse-power,** or the power that may be absorbed by a prony brake.

Indicated Horse-power. — The power that is actually generated in the cylinder of an engine is greater than the power delivered by the engine to do useful work. The ratio of these two powers is called the mechanical efficiency of the engine. Thus,

$$\frac{\text{Brake horse-power}}{\text{Indicated horse-power}} = \text{mechanical efficiency.}$$

The difference is the power absorbed in friction, in opening valves, operating governing mechanism, oil and fuel pumps, igniters, etc. The frictional horse-power is often taken as the indicated horse-power at no load. This is not quite true as the friction work of bearings, crosshead slides, pumps, etc., will be greater at full loads than at lighter loads.

The **mechanical efficiency** of an engine is zero at no load and increases to a maximum at the maximum horse-power. On account of the great number of auxiliaries, the mechanical efficiency of a gas-engine is lower than that of a steam-engine. While the efficiency of a steam-engine may be as high as 95 per cent, that of a gas-engine seldom runs above 90 per cent at full load.

It is clear that the more power that is required to take care of the auxiliaries of a gas-engine, the less will be the mechanical efficiency. In a Diesel engine the power required for the air compressor which furnishes the high-pressure air for the fuel injection must be charged to the engine. This decreases the useful

work for a given indicated horse-power and cuts down the efficiency in proportion.

Likewise, the power required to do the scavenging on a two-cycle engine cuts down the useful output and hence the mechanical efficiency.

Measuring the Output. — The output of a gas-engine, that is the brake horse-power, may be measured in many ways. For small engines, up to 200 horse-power, the best way is to use a prony brake for comparative or approximate results. The prony brake is not particularly accurate, neither is it suitable for large powers. In some cases it is impossible to get a suitable brake pulley on the engine shaft, in which case some other means must be used to measure the output of the engine.

When an engine is built to be direct connected to an electric generator, the best method of measuring the engine output is to measure the electrical output and correct it for the efficiency of the generator, thus obtaining the brake horse-power of the engine. The efficiency of an electrical generator may be secured more accurately than the readings and measurements of a prony brake, so that in general the latter method is much to be preferred rather than the former.

Where engines are tested by manufacturers, carefully calibrated generators are coupled up to the engines and the efficiencies found for different loads. Or, in many cases, the engines are connected to fans or blowers instead of electric generators. Either method gives a means of varying the output over a wide range and keeping the load steady at each point, a difficult thing to do with the prony brake.

The Prony Brake. — Prony brakes are made in a variety of forms, one of which is shown in Fig. 72. The brake wheel *W* should be made with flanges on the inside of the rim to retain water. The band *B* is best when made of a woven belting. This band is riveted to a plate at either end, the plates being forged to the adjusting screws *S*. The arm *A* has attached to it a block *E* cut out to the radius of the wheel. The adjusting nut *D* is most convenient when made in the form of a hand wheel.

Power is absorbed by friction of the band on the wheel, and in order to prevent burning, this band should be continually saturated with lubricating oil. The heat generated is carried away by the vaporization of the water in the wheel.

With a brake of this type, having a band three inches wide, the author has had no trouble in absorbing 15 horse-power where the peripheral speed was as high as 4800 feet per minute.

The horse-power of a prony brake is found by the equation

$$\text{Brake horse-power} = \frac{2\,\pi FRN}{33{,}000}.$$

Since the denominator is in foot-pounds per minute, it is obvious that the units in the numerator must be selected to give foot-pounds per minute in the numerator. Since the distance must be in feet, R must be in feet, giving as the distance $2\,\pi R$ for one revolution, or $2\,\pi RN$ as the distance moved through per minute. If the force F, measured on the scales, is in pounds, then $2\,\pi FRN$ will be in foot-pounds per minute as required.

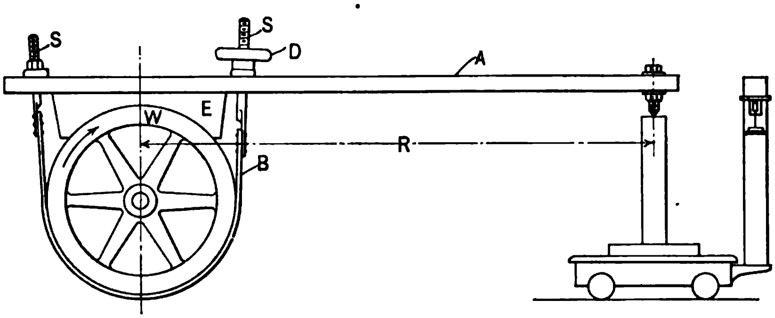


FIG. 72. — Prony Brake.

It is sometimes difficult for any one not familiar with the prony brake to thoroughly understand that, even though the force at the scales does not move we get the same effect as if it did, since the movement occurs at the periphery of the wheel. The force if applied at the rim of the wheel would have to be greater than the force at the radius R by the ratio $\frac{R}{\text{rad. wheel}}$. This would give the same work per minute as the distance would be decreased by the ratio $\frac{\text{rad. wheel}}{R}$.

Another way to look at it is to assume that the band is tight on the brake wheel and that the force F moves in a circle of radius R. Then, clearly, the distance moved through is $2\,\pi RN$ feet per minute.

Another point which is necessary to keep in mind is that this same formula is applicable to a prony brake wherever used, and that the type of gas-engine makes no change in the formula. This, we will see later, is not true of the formula for the indicated horse-power.

The factor F in the brake formula is the net load at R. The zero, or non-effective load, may be found by turning the engine slowly forward with the band loosed, and taking the weight; then in the opposite direction, taking the weight. The mean of these two weights is the zero load, and should be subtracted from the scale load in order to get the value F.

The Alden brake is one in which the rubbing surfaces are discs separated by a film of oil, and the heat is absorbed by water under pressure, which produces the friction. It is constructed by fastening a disc of cast-iron to the power shaft; this disc revolves between two sheets of thin copper joined at their outer edges, from which it is separated by a bath of oil. Outside the copper sheets on either side is a chamber which is connected to the water supply. The water flows through this chamber constantly, maintaining a uniform temperature. Any pressure in the chamber causes the discs to press against the revolving plate, producing friction which tends to turn the copper discs. As these are rigidly connected to the outside cast-iron casing and brake arm, the turning effect can be balanced and measured the same as in an ordinary prony brake. The pressure of the water is automatically regulated by a valve, which is partially closed if the brake arm rises above the horizontal and is partially opened if the arm falls below the horizontal. This brake with a constant head of water gives very close regulation. Another advantage is that it can be used for large powers.

Indicated Horse-power. — Since it is necessary to find the work done in foot-pounds per minute in order to find the horse-power of an engine, it is necessary to find the mean force acting on the piston and multiply this by the distance moved through per minute. The mean force is found by taking an indicator card from the engine cylinder. This indicator card is nothing more nor less than a graphical record of the pressure in the cylinder at all points of the cycle, but, knowing the laws of gases, many interesting facts may be read "between the lines."

The ordinates of the indicator card represent unit pressures on the piston when the length of the card represents the volume dis-

placed per stroke of the engine or the ordinates represent total pressures on the piston when the length of the card represents the stroke of the engine in feet. Thus in either case the area of the indicator card represents the work done in foot-pounds per working stroke. If the mean total pressure on the piston is computed by multiplying the mean height of the card in pounds per square inch P by the area of the piston in square inches A and if this total pressure is multiplied by the length of the stroke in feet L the result will be the foot-pounds of work done in the cylinder per cycle since there is one working stroke for each cycle. Since the horsepower of any engine is the foot-pounds of work done per minute divided by 33,000, the general horsepower formula may be written

$$\text{I.H.P.} = \frac{PLAN'}{33{,}000} \tag{1}$$

which is true for any type of gas- or steam-engine where N' is the number of cycles per minute. In a double-acting, single-cylinder steam-engine there are two cycles per revolution or $2\,N$ cycles per minute where N is the number of revolutions per minute. The horsepower formula for that engine then becomes

$$\text{I.H.P.} = \frac{2\,PLAN}{33{,}000}. \tag{2}$$

For a two-cycle, double-acting, single-cylinder gas-engine the same formula would apply since there are two cycles per revolution as in the steam-engine. If the engine is a four-cycle, double-acting, single-cylinder gas-engine formula (1) becomes

$$\text{I.H.P.} = \frac{PLAN}{33{,}000}, \tag{3}$$

since there is a cycle for each revolution.

Investigation of formulas (1), (2) and (3) will show that to change (1) into a form involving N, the revolutions per minute for any type of engine, it is only necessary to multiply (1) by the number of cycles per revolution and substitute N for N'. Thus in a four-cycle, single-acting, single-cylinder gas-engine there are two revolutions per cycle or $\frac{1}{2}$ cycle per revolution and the horsepower is

$$\text{I.H.P.} = \frac{PLAN}{2 \times 33{,}000}. \tag{4}$$

This suggests a general formula for use with single-acting four-cycle engines with any number of cylinders, n,

$$\text{I.H.P.} = \frac{n\,PLAN}{2 \times 33{,}000}. \tag{5}$$

Example: As a practical example in the use of the above formula, the author quotes here from a test made by him on a two-cylinder, double-acting, two-cycle engine. The motor cylinders had a diameter of $24\frac{3}{4}$ inches and a stroke of $43\frac{3}{8}$ inches. The average mean effective pressure for the four combustion chambers was 62 pounds per square inch. The net area of piston, allowing for rods, was 453 square inches and the revolutions per minute 103.23.

The horse-power of the engine then becomes

$$\text{I.H.P.} = \frac{4 \times 62 \times 43.375 \times 453 \times 103.23}{33{,}000 \times 12} = 1270.$$

This indicated horse-power included, in this two-cycle engine, the work done by the gas and air pumps, which amounted to about 262 indicated horse-power, leaving a net indicated horse-power of 1018. The output of the generator was 600 kilowatts, kept constant by means of a water rheostat. With a generator efficiency of 93 per cent, the brake horse-power of the engine would be

$$\frac{600}{.93 \times 0.746} = 864.$$

The mechanical efficiency of this engine was then

$$\frac{864}{1270} = 0.68 \text{ or } 68 \text{ per cent.}$$

Thermal Efficiency. — The mechanical efficiency of a gas-engine has been explained and illustrated by the above example. There is still another efficiency which is of more importance than the mechanical efficiency, the **thermal,** or heat efficiency. In order to understand this term we must first look at the relation of heat and work. The unit of heat, **British thermal unit,** has already been defined as the amount of heat required to raise the temperature of 1 pound of water from 62° to 63° F. Since heat can be transformed into work and work into heat, there must be a known relation existing between the two. That relation is called the mechanical equivalent of heat, and is 778 foot-pounds per B.t.u. This is usually denoted by J.

In a perfect engine, then, for every B.t.u. put into the engine we should take out 778 foot-pounds of work. Since there are 33,000 foot-pounds per minute in a horse-power, the heat to be supplied, theoretically, per horse-power, per minute is

$$\frac{33{,}000}{778} = 42.42.$$

Another value that is used and referred to frequently is the heat per horse-power hour, which is $60 \times 42.42 = 2545$.

Since the efficiency of a gas-engine is never perfect, or equal to 1, that is since we can never get 778 foot-pounds of work for each B.t.u. put in, we must put in more than 2545 B.t.u. in order to take out one horse-power for a period of one hour.

The general expression for any efficiency is

$$\frac{\text{Output}}{\text{Input}} = \text{efficiency.}$$

In any engine, then, to find the thermal efficiency, it is only necessary to find the heat actually supplied per horse-power and divide it into the output, **per horse-power hour,** which is always 2545 B.t.u.

The heat input must be found by measuring the fuel delivered to the engine per horse-power per hour, and multiplying this value by the heating value per unit of the fuel. The fuel may be based on the indicated horse-power or the brake horse-power, giving us two values of thermal efficiency. The efficiency based on the indicated horse-power is used frequently in design but that based on the brake horse-power, or output, is the one that tells the story of an economical or a wasteful engine. A purchaser of an engine is interested in knowing how much power he is going to have available for actual use, per ton of coal or per 1000 cubic feet of gas as the case may be. He does not care what percentage of heat will be transformed into work in the engine cylinder but what percentage will be transformed into work at the switchboard or at the belt drive.

The ratio between thermal efficiency based on brake horse-power and that based on indicated horse-power is the mechanical efficiency, or

$$\frac{\text{Efficiency based on B.H.P.}}{\text{Efficiency based on I.H.P.}} = \text{mechanical efficiency.}$$

To illustrate the method of finding the efficiency of a gas-engine, the author gives here the results of the gas-engine test referred to above. The fuel was blast-furnace gas, measured by means of a Venturi meter. It was found that the consumption of gas per hour during the 600-kilowatt test was 129,510 cubic feet at standard conditions. Since the indicated horse-power was 1270, the fuel consumption per I.H.P. hour was 102 cubic feet. The heating

value of the gas at standard conditions was 94.59 B.t.u. per cubic foot. The efficiency based on indicated horse-power was then

$$\frac{2545}{102 \times 94.59} = 0.264 \text{ or } 26.4 \text{ per cent.}$$

Since the brake horse-power was 864, the fuel consumption was 150 cubic feet per brake horse-power hour, giving an efficiency based on brake horse-power of

$$\frac{2545}{150 \times 94.59} = 0.1793 \text{ or } 17.93 \text{ per cent.}$$

It will be noted that $\frac{0.1793}{0.264} = 0.68$, the mechanical efficiency found above.

Maximum Efficiency. — The maximum efficiency of a gas-engine should be at about the rated load of the engine. The efficiency varies greatly with the load and there is scarcely any "flat" part of the curve, as shown by Fig. 73. This is an efficiency curve of a natural gas-engine rated at 200 brake horse-power. It will be noted that the efficiency drops off rapidly with the power, being 30.5 per cent at full load, 27 per cent at $\frac{3}{4}$ load, 21.74 per cent at $\frac{1}{2}$ load and 15 per cent at $\frac{1}{4}$ load. The dropping off of efficiency is inherent in gas-engines and this particular case is better than the average.

Maximum Power. — It will be noticed that the efficiency curve in Fig. 73 does not cover a range much above the rated horse-power. It is impossible to run a gas-engine at a power much greater than its rated power. It has been seen in an earlier chapter that the size of the cylinder is determined by the amount of fuel and air to be drawn into that cylinder, the amount drawn in being just enough to furnish heat to produce the desired power with an allowance for overload. If that allowance is 15 per cent, then 15 per cent is the maximum overload obtainable. It might be asked, why not allow more, say 50 per cent for overload. If that is done, then the rating of the engine is not honest, the engine will be rated too low, based on maximum efficiency, or, in other words, efficiency will be low at the rated power.

In a high-pressure oil-engine, similar to the Diesel, greater overloads are possible as fuel may be injected for a greater portion of the stroke than is done at the rated load, giving the same effect as increasing the cut-off of a steam-engine. This, however, is done

at a sacrifice of efficiency as the oil that is introduced into the cylinder late is only partly burned.

Efficiency at Various Compressions. — It was brought out in Chapters IV and VIII that the efficiency of the Otto cycle increased with increased compression. The theoretical efficiency is $\frac{T_1 - T_2}{T_1}$, where T_1 is the absolute temperature at the end and T_2 the corresponding temperature at the beginning of compression. This appears at first sight to be an excellent way of increasing the efficiency, but this trouble comes up; gases rich in hydrocarbons

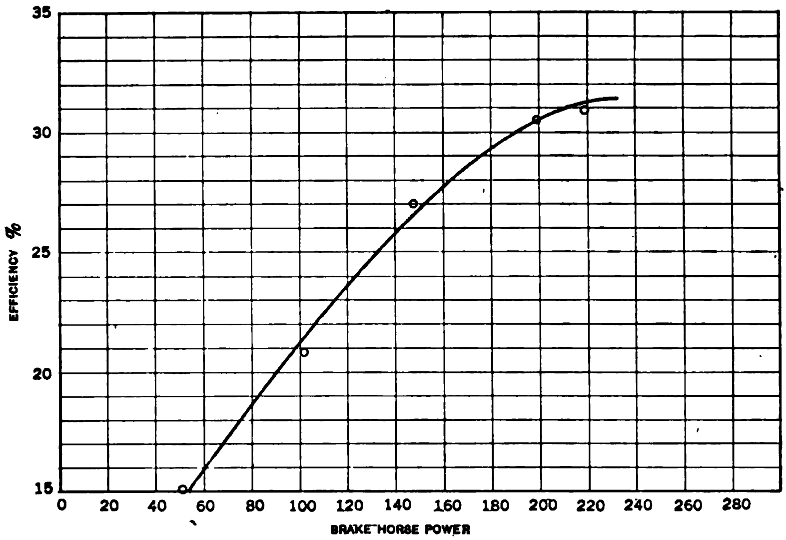


FIG. 73. — Efficiency Curve, 200 Horse-power Vertical Engine Running on Natural Gas.

have a low ignition temperature and if the compression is carried high enough, the resulting temperature will cause pre-ignition. Natural gas, being rich in hydrogen or methane, will withstand only low compression without pre-ignition, while some of the other leaner gases, such as blast-furnace and anthracite producer gases, will allow higher compression and consequently they are more efficient fuels.

Proportions of Cylinders. — The ratio of the length of stroke to diameter of cylinder is an arbitrary one, and a little investigation will show that this ratio may vary from 1 to 2. As a rule

the higher the engine speed, the nearer to 1 should be the ratio. Automobile engines which run at high speed are often made "square," or the diameter of the piston is equal to the length of the stroke. Recently the ratio has been increased in this line of work until in some engines it is as high as 1.5.

The condition which influences this ratio more than anything else is the shape of the combustion chamber. The most effective and efficient combustion chamber is one in which the ratio of enclosing area to volume is small. If the combustion chamber is cylindrical and the length equal to the diameter, then the ratio of area to volume is 6. If the length be equal to ½ the diameter, then the ratio is 8, but if the length be equal to ¼ the diameter, the ratio will be 12. If we assume the combustion chamber volume, or, what is the same thing, the clearance volume, to be 25 per cent, corresponding to about 100 pounds compression, the stroke of the engine will have to be four times the diameter of the piston to give a length of clearance volume equal to the diameter. This ratio of stroke to diameter is absurd, of course. If the ratio of length of clearance to diameter is reduced to ½, then the stroke of the engine will be twice the cylinder diameter, a ratio that might be taken as the maximum.

Working from the ratio of stroke to diameter we may find the value of the ratio of length of combustion chamber to diameter. Assuming a ratio of stroke to diameter of 1.5, we see that with 25 per cent clearance, the length of combustion chamber would be ⅜ of the diameter. Some engines are built with a ratio of length to diameter of combustion chamber of ½ without considering the ratio of stroke to diameter.

In double-acting engines the combustion space is so irregular in form that any assumption of ratio as above is of little value. In high-compression engines, such as the Diesel and semi-Diesel type, the combustion space is often made of smaller diameter than the cylinder proper, in order to get good proportions and at the same time have space for the insertion of valves.

Speed of Engines. — The speed of gas-engines, that is, the revolutions per minute, is governed by the character of the work to be done or by the piston speed. Automobile motors run at extremely high speeds in order to secure the maximum power per unit of weight. The maximum speed of such motors may be taken at 1800 revolutions per minute. A six-inch stroke engine

at this speed would have a piston speed of 1800 feet per minute. The rated horse-power of a four-cylinder, four-cycle automobile engine with six-inch cylinders would be

$$\text{H.P.} = \frac{6^2 \times 4}{2.5} = 57.5.$$

Since this formula,

$$\text{H.P.} = \frac{D^2 \times \text{no. cylinders}}{2.5},$$

is based on 1000 feet piston speed, an engine with a six-inch diameter cylinder and a six-inch stroke at 1800 revolutions per minute might be rated by the manufacturer at $1.8 \times 57.5 = 103.5$ horse-power.

The piston speed of stationary engines varies from 400 up to 1000 feet per minute. The gas-engine is inherently a high-speed engine on account of rapid combustion of the gases, but high speed means rapid wear, and commercially it is better to strike a happy medium around 600 to 800 feet per minute.

Marine engines should be designed to give proper propeller speeds. The maximum efficiency of modern propellers is around 100 revolutions per minute, but there are cases where they are run much higher with a corresponding loss in efficiency. They are seldom run at lower speeds.

Scavenging. — One disadvantage in running a gas-engine at high speed is the effect on the scavenging which is the process of cleansing the cylinder of burned gases. High piston speed means high velocity of gas through the valves, which in turn results in great fluid friction. In a four-cycle engine rarefied suction charge and high exhaust pressures will be the outcome, both tending to reduce the mean effective pressure, so that the increase in horse-power is not proportional to speed.

In two-cycle engines the effect of high speed is still more injurious. When scavenging is done by the crank-case compression, high speed necessitates larger passages from the crank case to the cylinder than can be conveniently used. If mechanically operated scavenging valves are used, these valves must be large in order to get the proper amount of scavenging air or mixture through them in the short time available. Since on a two-cycle engine these valves must be opened and closed every revolution, high speed is prohibitive on account of trouble with the heavy valve gear at

such speeds. Thus, as far as the valve gear is concerned, a four-cycle engine may operate at twice the speed of a two-cycle.

Perfect scavenging in a two-cycle engine is seldom accomplished. The best results are obtained when the scavenging air or mixture enters the cylinder at low pressure, not more than 6 pounds above the atmosphere, and in such a way that no eddy currents are formed. The entering air should drive the burned charge ahead of it in a path as straight as it is possible to make it. Any turn in the current of air, or any protuberance in the cylinder, causes the air and burned charge to mix and defeats the purpose of the scavenging air. For this reason, mechanically operated valves in the head of the cylinder are more efficient than scavenging ports uncovered by the piston.

EXERCISES

1. (*a*) Find the B.H.P. of an engine which has an I.H.P of 168, mechanical efficiency 83.5 per cent. (*b*) Find the I.H.P. of an engine with the same efficiency if the B.H.P. is 247.

2. On a prony brake test the total load measured on the scales was 436 pounds, the radius of the brake was 4 feet, speed 300 r.p.m. Find the B.H.P. if the weight on scales with band loose was 28 pounds when wheel was turned forward, and 18 pounds when wheel was turned in the opposite direction.

3. Find the I.H.P. of a double-acting, single-tandem, four-cycle gas-engine, 24 by 36 inches, running 125 r.p.m. M.e.p. = 78 lbs. per sq. in.

4. Find the I.H.P. of a two-cycle, single-cylinder, single-acting engine, 8 by 12 inches, running 350 r.p.m. M.e.p. = 64 lbs. per sq. in.

5. A test made on an engine using illuminating gas as fuel showed that when the I.H.P. was 26.4 the gas consumption was 370 cu. ft. per hour, heating value 630 B.t.u. per cu. ft. Find the thermal efficiency based on I.H.P. and on B.H.P. if the mechanical efficiency is 85 per cent.

6. Find the horse-power of a six-cylinder automobile motor with cylinders $3\frac{3}{4}''$ dia., $6''$ stroke. (*a*) At 1000 ft. piston speed. (*b*) At 1435 r.p.m.

CHAPTER XIV

THE COST OF POWER GENERATED BY INTERNAL COMBUSTION ENGINES

Plants of Various Sizes — Gas-, Gasoline- and Oil-engine Plants

General. — It is difficult to analyze the cost of power generated in a small plant, and arrive at a conclusion that is very near the truth, unless every condition is known beforehand. Since conditions vary with every plant, it is obviously impossible to make a detailed estimate, or an accurate estimate that will cover all small plants or all large plants. Certain assumptions may be made and on these assumptions certain estimates may be made, but the results thus obtained should be used with caution.

In order to show how estimates may be made, the author will discuss, in this chapter, certain costs that occur in small plants of different capacities. The discussion that follows will treat of isolated plants and the cost of power generated for a specific purpose, not for general distribution. Three types of engines will be taken up, steam, gas and oil. Steam-turbines will not be touched upon as that would involve problems concerned only with the relative merits of the turbine and the steam-engine.

The form of power to be generated is of importance. In order to have uniformity, we will assume that in each case the engine is direct connected to a direct-current generator, the current to be used for lighting and power in a manufacturing plant, during a working day of 10 hours, 6 days per week, or 300 days per year. The sizes of the plants selected will be 20, 100, 250 and 500 kilowatts. In no case will the cost of the land or building be included, as it is impossible to get even an approximate figure that would be of value for this item.

In general, the costs of the machines were secured from manufacturers. The prices in each case represent a mean of those submitted, in some cases, five or six, in others only two. The cost of the foundation in each case was based on the floor space,

and of the piping, on the cost of piping in similar plants. The cost of the machines in each case includes erecting, freight for three or four hundred miles, cartage, etc.

The "fixed charges" were apportioned for the steam and producer gas plants as follows:

	Per cent
Interest on investment	5
Depreciation	5
Repairs	2
Taxes	1
Insurance	1
Total	14

The depreciation will not be the same on all parts of the plant, nor will the cost of repairs be uniform over the whole plant, but such figures as were selected represent the average depreciation and repairs on the whole plant. For the high-pressure oil-engines, the depreciation will be assumed to be 5½ per cent and the repairs 2½ per cent, for it has been the experience that these plants cost more to keep in repair than plants where working pressures are not so high.

Since no plant can be expected to run at full load all the time, we will assume a load factor of 75 per cent. This factor would be high for a plant selling power, in fact, it would be a maximum for the type of plant under consideration.

The wages of a first-class engineer for the larger plants was assumed to be $4 per day, firemen $2.50 per day, and for second-class engineers, for the smaller plants, $2.50 per day. In the smaller plants and in all oil-engine plants, there was no allowance made for a night man.

The coal was based on 13,000 and oil on 19,000 B.t.u. per pound. The price of water was assumed to be 10 cents per 1000 gallons, lubricating oil 22 cents per gallon and cylinder oil 30 cents per gallon.

Twenty-kilowatt Plant. — The approximate costs of three different types of 20-kilowatt plants are given below. For the steam plant the cost is based on a high-speed, horizontal engine, running non-condensing. The type of boiler selected was the vertical fire-tube on account of the low cost, being $125 less than the horizontal return-tubular type.

The coal consumption was based on, first, efficiency of boiler, 60 per cent; and second, steam consumption of engine of 35 pounds

per indicated horse-power per hour. An additional 15 per cent was allowed on the coal for banking and losses.

The water cost was based on the above, with an allowance of 15 per cent for leakage, etc.

For the gas-engine plant, an anthracite producer is selected as being the best adapted for the work. The coal cost is based on 2 pounds per horse-power hour including stand-by losses. The care of a plant of this size will not require the full time of one man; in fact less than half of it; hence the labor cost of $300. The cost of water was based on 2 cubic feet per kilowatt hour for the entire plant, fresh water to be used continuously.

For the oil-engine plant a low-pressure oil-engine is selected as being better suited for this small plant than a high-pressure engine. The fuel cost is based on 1 pound of oil per brake horse-power hour, and the cost of water on a consumption of 5½ gallons per horse-power hour.

A table showing the itemized cost of each of the three particular types of plant follows:

COST OF POWER — 20-KILOWATT PLANT

	Steam plant.	Producer-gas plant.	Low-pressure oil-engine plant.
Brake horse-power	32	32	32
Indicated horse-power	35	....	....
Cost of engine and foundation	$800	$1200[1]	$1600[2]
Generator and switchboard	$600	$600	$600
Boiler, foundation, piping	$450	....	....
Producer and washer	....	$735	....
Total	$1850	$2535	$2200
Yearly cost:			
Coal at $4.00 per ton	$880	300	....
Oil (fuel) at 4 cents per gallon	....	....	400
Labor	750	300	200
Fixed charges	260	355	308
Oil, waste and supplies	125	135	125
Water	50	110	40
Total	$2065	$1200	$1073
Cost per kilowatt hour, cents	4.6	2.68	2.38

[1] Air compressor included in this item.
[2] Air compressor and oil tank included in this item.

In order to show the varying cost of power with different fuel values, power cost curves have been plotted with cost of fuel as abscissæ and cost in cents per kilowatt hour as ordinates. The

curves for the 20-kilowatt plant are shown in Fig. 74. The units on the horizontal scale are "dollars per ton" for the coal cost of the steam- and gas-engines, and "cents per gallon" for the oil-engine. The fuel cost for the steam-engine is a greater percentage of the total cost than for the gas-engine, hence the rapid rise in the line representing the steam-engine cost.

One-hundred-kilowatt Plant. — For the steam plant a high-speed, non-condensing engine is selected, as the first cost and fixed charges of a condensing engine of this size would be prohibitive. The boiler selected is of standard water-tube design. The coal and

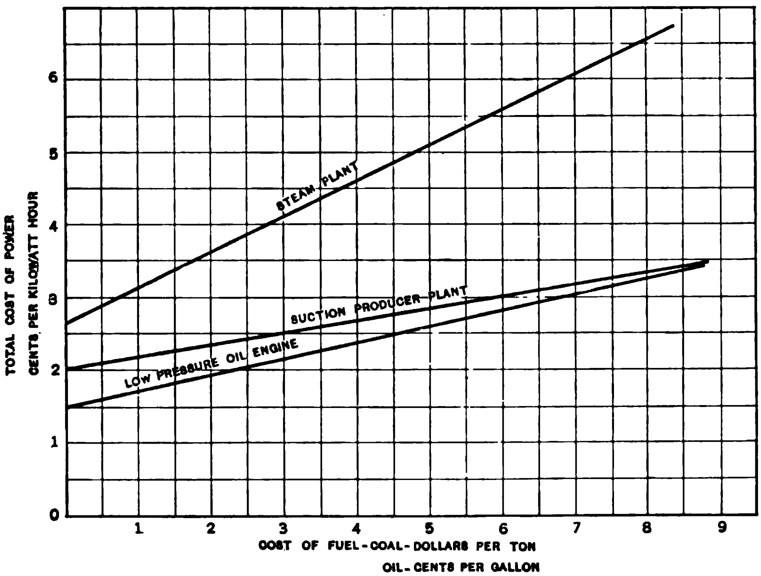


FIG. 74. — Cost of Power Generated by 20-kilowatt Plants.

water cost are based on boiler efficiency of 60 per cent, engine to use 30 pounds of steam per indicated horse-power per hour, with an additional allowance of 15 per cent for losses.

For the producer plant a suction producer operating on anthracite coal is assumed as the most convenient and the cheapest. A bituminous coal producer of this size would cost $3000 to install compared with $2200 for the anthracite producer. The engine cost is based on a three-cylinder, vertical unit. The coal cost is based on 1½ pounds per brake horse-power hour at 75 per cent load factor, 10 per cent being added for stand-by losses.

The water rate is assumed to be 15 gallons per horse-power hour, 5 for the engine and 10 for the producer and scrubber, using the water only once.

The 100-kilowatt oil-engine selected for use in this discussion is of the high-pressure type, either Diesel or semi-Diesel. The item of engine cost includes all auxiliaries, tanks, oil pump, air compressor, etc. The fuel cost is based on 0.55 pound per brake horse-power hour.

The comparison of the 100-kilowatt steam-, gas- and oil-engines appears in the following table:

Cost of Power — 100-kilowatt Plant

	Steam plant.	Producer-gas plant.	High-pressure oil-engine plant.
Brake horse-power	150	150	150
Cost of engine and foundation	$3000	$5700	$11,500
Boiler, pump and piping erected	4400	....	....
Producer and piping	....	2200	....
Generator and switchboard	1600	1600	1,600
Steel stack and foundation	500	....	....
Total	$9500	$9500	$13,100
Cost per year:			
Coal at $4.00 a ton	$3200	$1100	....
Fuel oil at 4 cents a gallon	....	....	$1,000
Attendance	1650	750	750
Fixed charges	1330	1330	1,965
Oil, waste and supplies	175	175	175
Water	150	500	170
Total	$6505	$3855	$4,060
Cost per kilowatt hour, cents	2.89	1.71	1.80

If a low-pressure engine had been used instead of a high-pressure, the cost of the plant would have been $4000 less. In that case the yearly fixed charges would have been 14 per cent on $9100, about $1275. The saving on this item would thus be $685 per year. The oil consumption for the low-pressure engine would be about 46,000 gallons, costing $1840 against $1000 for the high-pressure engine, the difference being $840 in favor of the high-pressure engine. The net difference is $155 in favor of the high-pressure engine. But if oil costs 3 cents per gallon, the balance would have been $55, and in favor of the low-pressure engine. This interesting point is brought out to show that as the price of fuel increases, it is a paying investment to put money into more efficient equipment.

In order to show the effect of increased price of fuel on power cost, Fig. 75 is shown here — a chart similar to the one in Fig. 74, except that it is for the 100-kilowatt plant.

Two-hundred-fifty-kilowatt Plant. — For this size plant the steam-engine is assumed to be a non-condensing, tandem-compound Corliss. The steam consumption of this engine would be about 22 pounds per horse-power hour. This figure was used to get the size of the boilers and the coal consumption. In working out the coal consumption, 10 per cent was allowed for auxiliaries and 10 per cent for stand-by losses.

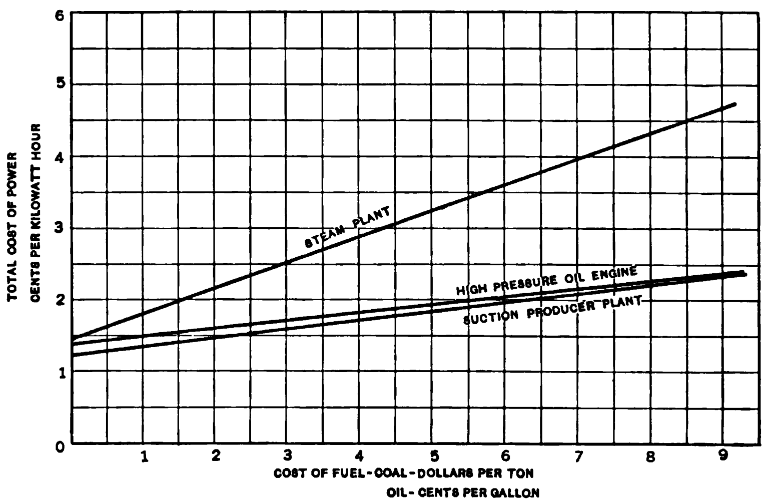


FIG. 75. — Cost of Power Generated by 100-kilowatt Plants.

The gas-engine used for the 250-kilowatt plant is of the four-cylinder vertical type, gas to be furnished by one anthracite producer. The coal cost is based on 1.5 pounds of coal per brake horse-power hour, including stand-by losses. The water charge is based on 7½ gallons per brake horse-power hour, using the scrubbing water over and over.

The oil-engine selected is of the high-pressure type, with a consumption of one-half pound of oil per brake horse-power hour.

It is interesting to note that the 250-kilowatt steam plant is the cheapest to operate with fuels at low cost. In Fig. 76 it will be noticed that the steam-engine line crosses the suction producer line when coal is worth $0.75 per ton, and the steam- and oil-engine

lines cross with coal at $1.50 per ton and oil at 1.50 cents per gallon. It is noticeable, also, that the line for the steam plant is less steep than for the 20- and 100-kilowatt plants. This is because the fuel cost is less in proportion to the total cost than in the smaller plants.

COST OF POWER — 250-KILOWATT PLANT

	Steam plant.	Producer-gas plant.	High-pressure oil-engine plant.
Brake horse-power	365	365	365
Cost of engine, foundation, etc.	$5,850	$13,000	$26,700
Boiler or producer and piping	5,600	5,125	....
Steel stack and flues	600	....	....
Generator, switchboard and wiring	4,000	4,000	4,000
	$16,050	$22,125	$30,700
Cost per year:			
Coal at $4.00 a ton	$6,000	$2,500	....
Oil at 4 cents a gallon	....	....	$2,280
Attendance	2,000	1,500	900
Fixed charges	2,240	3,100	4,600
Oil waste and supplies	225	225	225
Water	335	600	400
Total	$10,800	$7,925	$8,405
Cost per kilowatt hour, cents	1.92	1.40	1.50

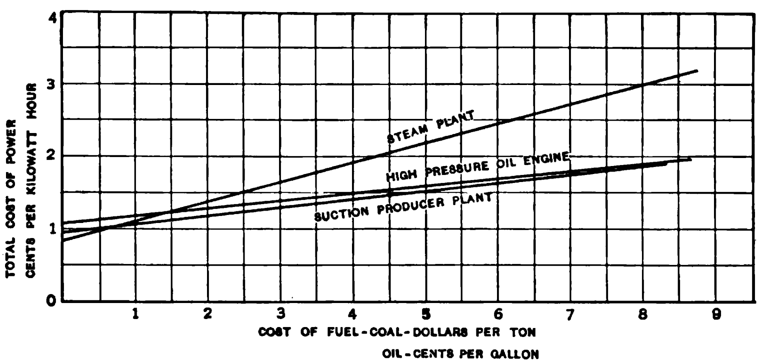


FIG. 76. — Cost of Power Generated by 250-kilowatt Plants.

Five-hundred-kilowatt Plant. — For the steam plant of this capacity a compound-condensing Corliss engine is assumed. The steam consumption would be about 18 pounds per indicated horse-power hour for the main engine, with 20 per cent additional for auxiliaries and stand-by losses. It is also assumed that condensing water can be had at no expense except for pumping, which is allowed for in the cost of the plant, and steam for auxiliaries.

For the producer plant a bituminous pressure producer is assumed for several reasons, one of which is that the size is large for a suction producer, and in this discussion we are limited to one unit in each case for the sake of uniformity. The bituminous producer is well adapted to this size of unit and it is well to point out the possibilities of such an installation. The cost is more than for a plain suction producer, but the efficiency is probably higher; certainly the pressure type is more flexible.

The coal cost is based on 1.25 pounds of coal per horse-power hour, including all stand-by losses. The water is based on 7½ gallons per horse-power hour for the engine, producer and scrubber. In this case the water for the scrubber will have to be used over and over. If this is not done the water consumption will be doubled. In order to make it possible to use this water over and over, it should be cooled by spray nozzles after being run into a settling tank where the dust from the scrubber is deposited. The cost of the tank, nozzles, etc., has been taken care of in the cost of producers and auxiliaries.

For the oil-engine the selection is a Diesel type, the cost including air compressor, oil tanks and pumps, piping, etc.

Cost of Power — 500-kilowatt Plant

	Steam plant.	Pressure producer gas plant.	High-pressure oil-engine plant.
Brake horse-power	725	725	725
Cost of engine, foundation, etc.	$9,300	$25,000	$50,500
Boilers and pumps	10,000	....	....
Stacks and flues	1,200	....	....
Condenser	2,000	....	....
Producer and auxiliaries	....	13,000	....
Generator, switchboard, etc.	9,500	9,500	9,500
Total	$32,000	$47,500	$60,000
Cost per year:			
Coal at $4.00 a ton	$10,000	$4,080	....
Fuel oil at 4 cents a gallon	....	....	$4,480
Attendance	2,600	2,600	1,200
Fixed charges	4,480	6,640	9,000
Oil, waste and supplies	400	400	400
Water	420	1,000	840
Total	$17,900	$14,720	$15,920
Cost per kilowatt hour, cents	1.59	1.31	1.42

The diagrams showing cost of power in the different 500-kilowatt plants with varying cost of fuel are given in Fig. 77.

Other Fuels. — While the foregoing tables and diagrams cover the cost of power generated by engines using the more common fuels, in some cases other fuels may be used for special reasons. Among these fuels are natural gas, illuminating gas, gasoline and kerosene.

In order to give some idea of the cost of power generated by these fuels, a specific plant, 100 kilowatts capacity, will be discussed here.

The cost of the two gaseous fuels is based on an efficiency of 25 per cent. For the natural gas plant this means a consumption of 11.3 cubic feet per horse-power hour with gas having a heating value of 900 B.t.u. per cubic foot. For illuminating gas the figures are 17 cubic feet with gas of 600 B.t.u. per cubic foot.

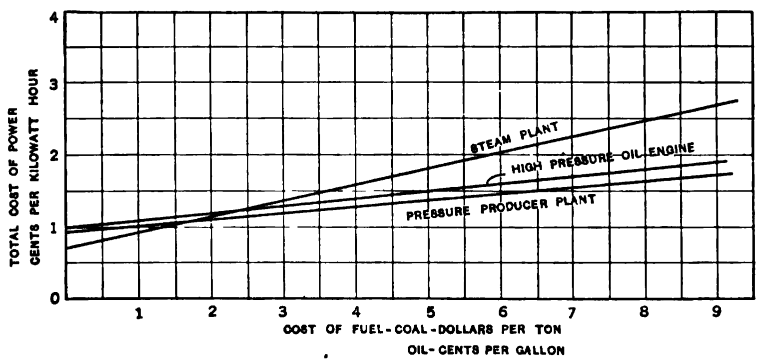


FIG. 77. — Cost of Power Generated by 500-kilowatt Plants.

The cost of gasoline is based on an efficiency of 20 per cent, assuming the gasoline to be used in a carbureter. The fuel consumption with carbureted kerosene would be practically the same.

The natural gas cost is based on 20 cents per thousand cubic feet, illuminating at 60 cents, and gasoline at 14 cents per gallon.

Figs. 78 and 79 illustrate the cost of power generated by natural gas and illuminating gas respectively, in various size units, these costs being exclusive of land and buildings. In many cases the cost of land and buildings does not amount to much for plants of this size, as the floor space required for vertical engines is small, and auxiliaries do not take up much room.

Up to 30 or 40 cents per thousand cubic feet, natural gas makes an ideal fuel for the gas-engine as it is clean, high in heating value, and the supply is usually reliable. In some districts the pressure

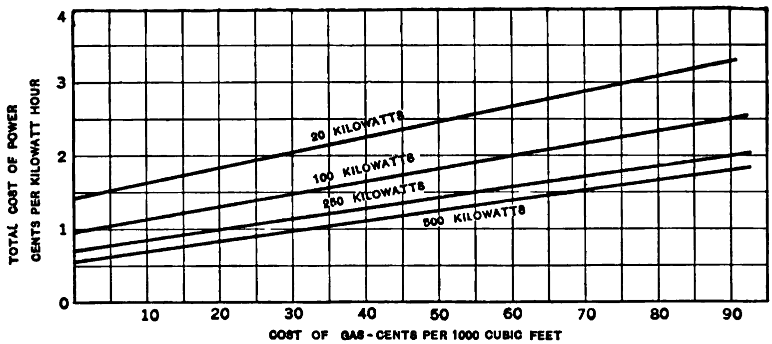


FIG. 78. — Cost of Power Generated by Plants Using Natural Gas for Fuel.

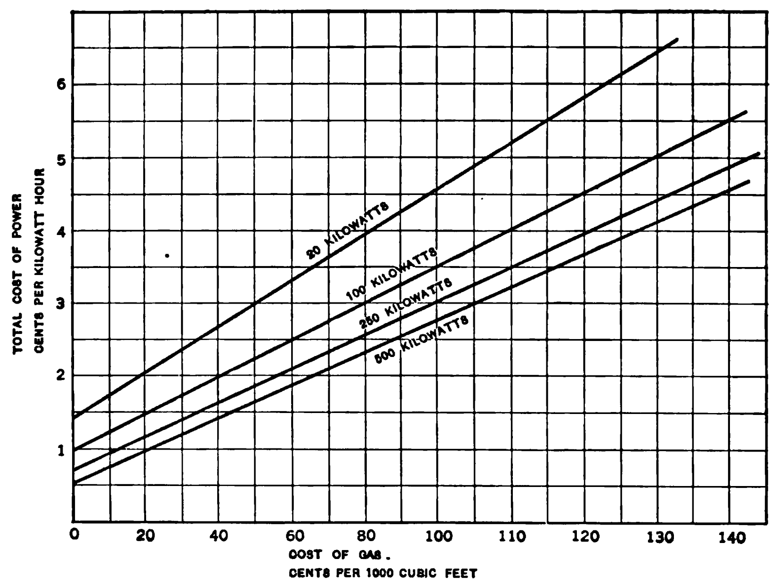


FIG. 79. — Cost of Power Generated by Plants Using Illuminating Gas for Fuel.

falls appreciably in severe weather in the winter, particularly in the early evening hours. In such localities provision must be made for other power at that time.

COST OF POWER — 100-KILOWATT PLANT

	Natural gas.	Illuminating gas.	Carbureted gasoline.
Brake horse-power	150	150	150
Cost of engine, foundation and piping	$6000	$6000	$6300
Generator, switchboard, etc	1600	1600	1600
Total	$7600	$7600	$7900
Cost per year:			
Gas or gasoline	$762	$3444	$5572
Attendance	750	750	750
Fixed charges	1064	1064	1106
Oil, waste and supplies	175	175	175
Water	170	170	170
Total	$2921	$5603	$7773
Cost per kilowatt hour, cents	1.30	2.49	3.45

Duplicate Units. — In each case of the foregoing discussion one unit was assumed to constitute the entire plant. There are many places where such an arrangement would be inadvisable; in fact, in the majority of cases it would be better to have two units instead of one. The reason for this is not particularly the unreliability of one unit, but if the load varies greatly at different periods of the day or at different seasons, it would pay to put in two or more units so that each might carry a large percentage of its rated capacity whenever it is in operation. The advantages and disadvantages of such an installation would have to be investigated for each individual plant and settled in a way best suited to meet local conditions. In general, it might be said that the low efficiency of gas-engines at light loads necessitates running each unit at a load near its rating in order to obtain the best results in fuel consumption. In plants where the power generated is to be used largely for lighting, or to operate non-reversing motors, it might be advisable to install alternating-current generators. In general this will lessen the cost of the generators, but add to the cost of switches and exciter equipment.

In order to give some idea of the cost of a plant with duplicate sets, generating alternating current, there follows an analysis of the cost of a typical suction producer plant as installed by the Foos Gas Engine Company for operating on anthracite coal. This plant consists of two 200-horse-power, three-cylinder vertical

engines direct connected to 60-cycle, three-phase, 2300-volt alternators operating in parallel. The cost of the plant was as follows:

Engines	$14,000
Generators	5,000
Exciters	500
Switchboard	1,000
Foundations	600
Piping and wiring	250
Producers	5,600
Auxiliaries and Installation	1,200
Total cost of plant	$28,150
Cost per year:	
Coal, 675 tons at $4.00	$2,700
Attendance	1,500
Fixed charges at 14 per cent	3,940
Oil, waste and supplies	225
Water	675
Total	$9,040

At 75 per cent load the cost per kilowatt hour would be 1.51 cents.

The coal cost for the above plant was based on 1.5 pounds per kilowatt hour at 75 per cent load, including stand-by losses 14 hours per day. At full load this plant would operate on 1.25 pounds. The water charge is based on 7½ gallons per horse-power hour at 75 per cent rating. Of this amount, 5½ gallons will be used by the engine, the balance in the producer and scrubber, the assumption being that the scrubber water is used continuously, this being provided for in the cost of the plant.

The following table gives the cost in dollars per kilowatt, not including land and buildings, of the plants discussed in this chapter.

Size of plant, kilowatts.	Steam.	Producer gas.	Oil.	Natural or illuminating gas.	Gasoline or kerosene.
20	92.50	126.75	110.00	92.50	110.00
100	95.00	95.00	131.00	76.00	79.00
250	62.50	88.50	122.80	70.00	....
300	....	94.00	....	....	....
500	64.00	95.00	120.00	70.00	....

There are apparent discrepancies in the above table which become clear on investigation. The increased cost of the 100-kilo-

watt over the 20-kilowatt steam plant is due to the fact that a water-tube boiler was assumed for the 100-kilowatt instead of the fire-tube boiler used in the 20-kilowatt plant. The increase is due to the greater cost of the chimney. The increase in cost of the 500-kilowatt steam plant over the 250-kilowatt is due to using a condensing engine in the larger plant.

The 300-kilowatt producer plant costs more than the 250-kilowatt because it is composed of two units instead of one. The 500-kilowatt producer plant is high on account of using a bituminous producer. If anthracite suction producers had been used in this plant, the cost would have been $88 per kilowatt. This is high relative to the 250-kilowatt plant because two or three producers would be needed for the larger plant; it would be impossible to get satisfactory results from one suction producer to supply a 500-kilowatt unit.

The 20-kilowatt oil-engine plant is low in cost because of the low use of a low-pressure engine, while the assumption for the 100-kilowatt plant was a high-pressure engine. If a high-pressure engine were used in the small plant the cost would have been $130 per kilowatt.

CHAPTER XV

TYPES OF ENGINES

Small Gas- and Gasoline-engines

General. — There are so many ways in which different types of internal combustion engines may be classified that it would be out of place for anyone to classify them in one way, and claim that way to be the best. In this and the two succeeding chapters the author wishes to illustrate and describe briefly some of the types of engines made in the United States and abroad. In order to avoid confusion a partial classification has been attempted. It is as follows:

1. Small gas- and gasoline-engines.
2. Large gas-engines.
3. Oil-engines.

There is no necessity of differentiating between gas- and gasoline-engines. With the exception of the carbureter and the mixing valve there is no inherent difference. In fact, the carbureter may be taken off a gasoline-engine and a mixing valve substituted, and the engine will work efficiently on any rich gas. Hence, the gasoline- and gas-engines are put under one classification..

The engines that are shown in the following chapters have not been selected for their superiority; they represent types of engines which are generally conceded to be good practice today, and which show the trend of the development for tomorrow.

SINGLE-CYLINDER ENGINES

Foos Single-cylinder Horizontal Type. — The engine illustrated in Fig. 80 is built in sizes from 3 to 90 horse-power. It is of the four-cycle type, built to operate on gas and liquid fuels, and is water cooled. The valves are attached to the sides of the cylinder and not to the head. They are both vertical and both mechanically operated from the bottom. Both valves are in separate detachable chambers surrounded by water-cooling jackets.

The **jacket wall** is cast solid with the cylinder. The water space is closed by the cylinder head and when the latter is removed an opening is exposed sufficiently large for the removal of any accumulation.

The **piston** is of the trunk type, made of close-grained cast-iron, and carries four rings.

Ignition is of the make-and-break system, with the wiper contact shown in Fig. 36. The gear shown between the spokes of the flywheel in Fig. 80 operates the igniter rod which is seen extending up to the inlet-valve chamber. The cams which operate the valves are on the same shaft with the gear, between the gear and the crank discs.

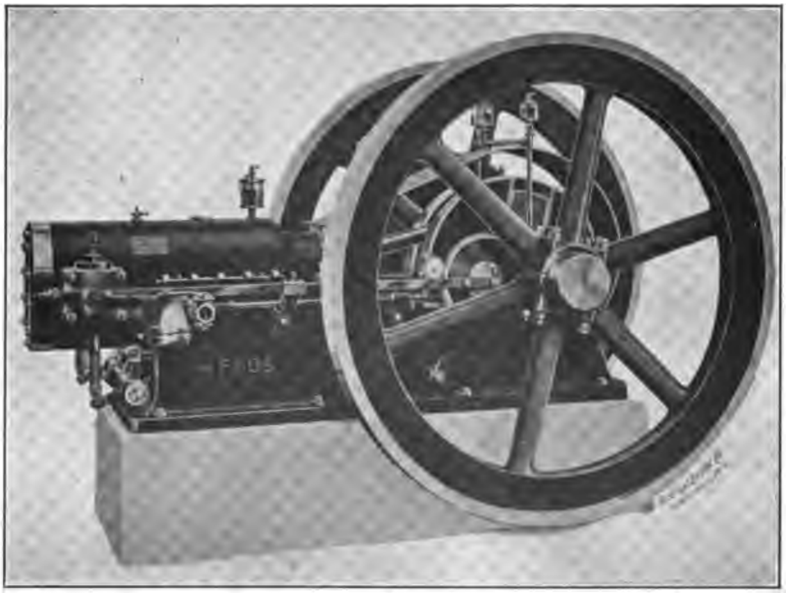


FIG. 80. — Foos Single-cylinder Engine.

The **governor** of this engine operates on the fuel-valve, either on the hit-and-miss or the throttling principle. In either case the governor does not interfere with the main inlet- or exhaust-valves. The governor proper is of the centrifugal fly-ball type.

The **shaft** is of the center crank type and is a solid forging. The counterweights are bolted to the crank-arms and for additional strength have a rim that extends entirely around the crank-arms.

The **connecting-rod** is of the marine type at both ends, is of circular section, and has phosphor bronze bearings.

The "New Way" Engine. — This engine, illustrated in Fig. 81, is of the air-cooled type, made by the "New Way" Motor Company, Lansing, Michigan. It is of the four-cycle type, built to use gasoline, and is made in 2½, 3½ and 4½ horse-power vertical

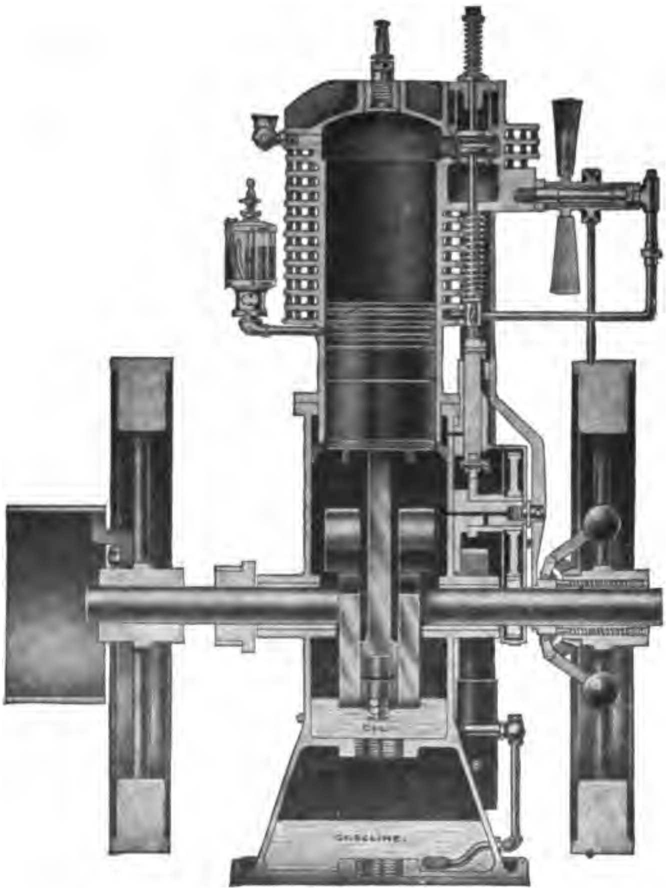


FIG. 81. — "New Way" Air-cooled Gasoline-engine.

units, and in a 6-horse-power horizontal unit. It is also made in 8 and 12 horse-power, 2-cylinder opposed units.

The engine base is made in two parts, an upper and lower half, hinged at the back. When it is desirable to take up the connecting-rod bearings, the cylinder and upper crank-case can be tipped back on the hinge and the working parts exposed to view.

The **carbureter** on this engine is of the Venturi type, in which the gasoline is drawn up from the base of the engine by the suction during the intake stroke. The *ignition* is of the high-tension, jump-spark system, current being furnished by a high-tension magneto.

The **cylinder** and **head** are cast integral and the valves are in cages, bolted into place. The exhaust-valve is operated by a cam, while the inlet-valve is automatic.

The **governor** works on the hit-and-miss principle and cuts off both the gas and the ignition on the "miss" strokes.

The Ferro Motor. — During the last three or four years a demand for motors to be used on row-boats has sprung up. These motors are built into one compact unit, that is, motor, propeller shaft, and gasoline tank are all fixed on the same frame. The motor is usually horizontal, with a vertical shaft. The frame brackets are clamped on the stern of the boat, with the shaft overhanging.

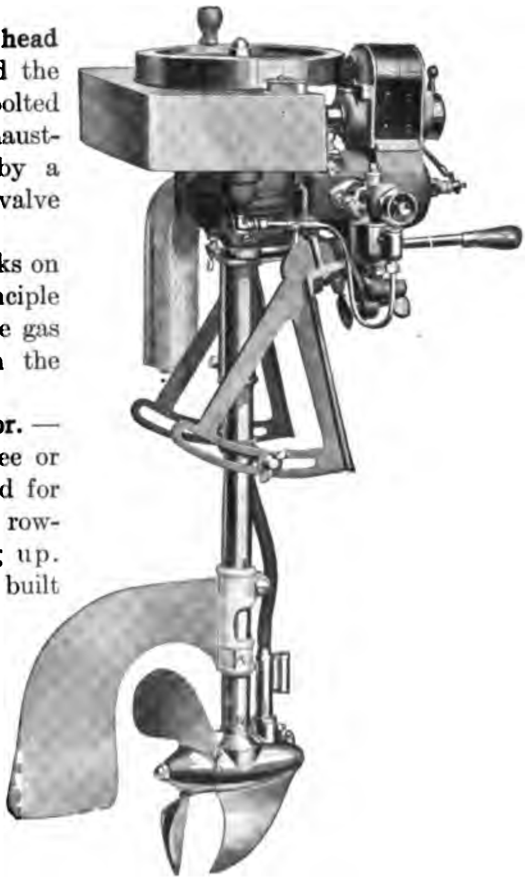


FIG. 82. — Ferro Gasoline Row-boat Motor.

One of these motors is illustrated in Fig. 82. It is of the single-cylinder, two-cycle reversible type. Reversing is done by timing the spark. The ignition system is Bosch high-tension, with reversible magneto. The carbureter is of the float-feed type. The handle shown at the right is for steering. This handle turns the propeller and rudder together. The engine is water-cooled, the

pump being down on the horizontal propeller shaft. The gasoline tank is shown in such a position that there is always a head of gasoline on the carbureter. The weight of the complete outfit is less than 100 pounds.

The Otto Engine. — The Otto engine shown in Fig. 83 is made in single-cylinder units in eight sizes from 40 to 170 horse-power, and in twin-cylinder units in eight sizes from 80 to 340 horse-power. The engine works on the four-stroke cycle.

The **frame** and **cylinder** are cast separate, and the frame extends forward so as to serve as a base for the cylinder and head. These parts simply rest on this base and are not fastened to it, thus permitting free expansion and contraction of the heated parts.

The **cylinder** and **jacket** are cast in one piece, with large water spaces. Separate cooling-water connections are provided for cylinder and head, so that water to these parts may be adjusted separately. A solid wall separates the cooling-water spaces in the cylinder and head, so that no cooling water passes through a packed joint.

The **piston** is of the trunk type and is made of the same grade cast-iron as the cylinder. The pin is held in place by set-screws and keys.

The **cylinder head** contains the combustion chamber. All valves, together with the igniter, are mounted on it. The entire head is water jacketed. The valve seats are removable and may be easily replaced in case of wear. The valve stems are provided with renewable bushings.

The **crank-shaft** is of 0.40 carbon open-hearth steel, forged from the billet and polished all over. The counterbalance weights are fastened to the crank-arms.

On single-cylinder commercial engines two **flywheels** are provided. On the direct-connected special electric type and on the twin-cylinder type, but one extra heavy flywheel is used.

The **connecting-rod** is of 0.20 carbon open-hearth steel. The boxes at each end are of the marine type.

Ignition is accomplished by means of an oscillating magneto, mounted on the cylinder head and positively driven. The movement of the armature in this magneto is limited to a few degrees. No spark coil is necessary. The igniter is of the make-and-break type, consisting of a flange in which are housed the fixed

FIG. 83. — Otto Single-cylinder Gas-engine.

and movable electrodes. An interrupter lever and spring complete the device. The igniter lever, operated by a crank disc on the gear shaft, oscillates the armature of the magneto and separates the electrodes at the proper time, causing a break in the circuit, thus producing the spark.

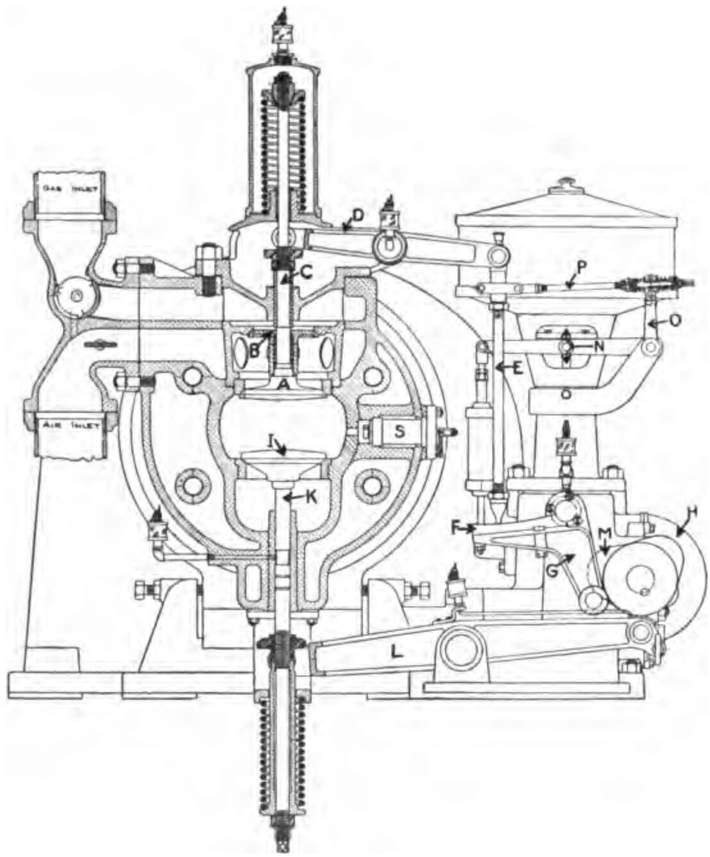


FIG. 84. — Otto Gas-engine — Section Through Combustion Chamber.

The **gear shaft** is driven from the crank-shaft by spiral gears running in oil. This shaft carries the spiral gears that drive the governor. On it are also mounted the cams that actuate the inlet- and exhaust-valves. The gear shaft is carried in bearings fastened to the frame at both ends, permitting the removal of the valves, cylinder head and cylinder without dismantling the

gear shaft, nor is the alignment of the gear shaft bearings affected by the expansion and contraction of the cylinder head or cylinder jacket.

Governing is accomplished by means of a high-speed ball type governor which controls the lift of the mechanically operated inlet-valve, regulating the amount of the mixture of air and gas according to the load. In Fig. 84 is shown a section through the valves and combustion chamber. It will be noticed that the charge enters the cylinder through two separate ports, controlled by hand-operated throttles and the poppet inlet-valve *A*. The poppet valve *B*, mounted flexibly on the valve-stem *C*, times the admission of the gas in relation to the air inlet. The inlet-cam *H*, acting upon the lever *G*, upon which the steel shoe *F* is mounted, opens the inlet-valve *A* in a positive manner by means of the push rod *E* and the valve lever *D*. The lift of the valve *A* is decreased or increased when the pivot pin *N* is raised or lowered by the governor, thus causing the lower end of the push rod *E* to be moved by means of lever *O* and rod *P* close to or farther away from the fulcrum of the lever *G*. There is always a slight amount of play between the end of the rod *E* and the steel shoe *F*, so that the work performed by the governor in swinging the rod *E* is very light. A dash-pot consisting of a cylinder and piston, oil cushioned and connected to the governor lever, steadies the operation of the governor and prevents sudden variation of speed under fluctuating loads.

The **exhaust-valve** *I* is operated by the cam *M* through the lever *L*. *S* is the igniter which was referred to in a previous paragraph.

The "Ingeco" Engine. — The four-cycle engine illustrated in Figs. 84a and 84b is built by the International Gas Engine Company, Cudahy, Wisconsin, in 12 sizes from 6 to 60 horse-power. The fuels used are gas, gasoline, kerosene or any other oil of 39° Baumé or lighter. When the heavier fuels are used, the engine is started on gasoline and changed over to the heavier fuel after becoming warmed up.

The **main-frame** of the engine is cast separate from the cylinder which is bolted to the frame. Reservoirs are cast under the main bearings and contain a supply of oil which is carried up onto the shaft by means of chains. The **bearings** are set at an angle of 45 degrees and are lined with removable babbitted

shells. The frame proper rests on and is bolted to a cast-iron sub-base.

The **cylinder-head** contains the **inlet-** and **exhaust-valve cages** at top and bottom, and the igniter in the center of the end. The valve cages may be removed easily for inspection and regrinding of the valves. The **water-jacket** of the head is independent of the cylinder jacket, the water being controlled by a separate valve.

The **piston** is of the regular trunk type with from three to five

Fig. 84a. — Valve Gear of Ingeco Gas-engine.

rings, depending on the size of the engine. The **piston-pin** is hollow and is secured in the piston by two set-screws.

The **crank-shaft** is forged from a solid, open-hearth billet, the arms being slotted out and finished all over. The spiral gear for the cam-shaft drive is pressed on and keyed with a tapered key.

The **connecting-rod** has a bronze piston-pin bearing adjusted with wedges, and a babbitt-lined crank-pin bearing which is adjusted by means of liners or shims. The piston end of the rod is of the closed type, the crank end of the marine type.

The **cam-shaft** has three bearings, one on the main-frame outside the crank-shaft, the other two at the operating end, one on either side of the cams. The spiral gear, which drives the cam-shaft, runs in oil.

The **cams** and **valve-gear** are shown in detail in Fig. 84a. The cams are keyed to the cam-shaft by means of straight keys. The **exhaust-cam** operates the exhaust-valve through a rocker arm of *I* section. The **inlet-cam** opens the suction-valve by means of the vertical push rod and rocker arm. This push rod

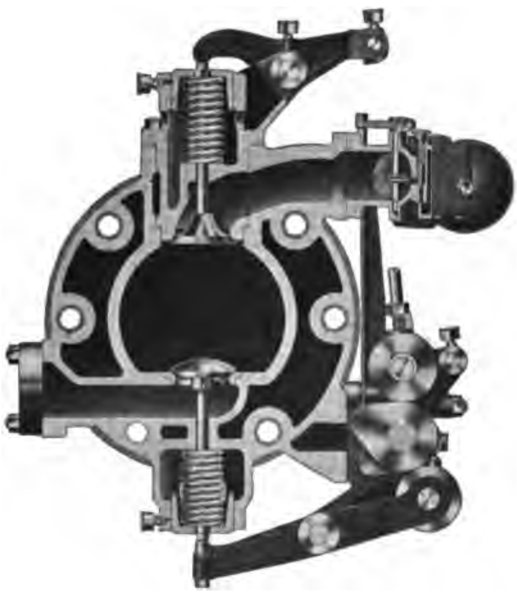


Fig. 84b. — Section Through Combustion Chamber of "Ingeco" Gas-engine.

has a right- and left-hand thread for adjustment. A **relief-cam** opens the exhaust-valve during part of the compression stroke, thus relieving compression to a certain extent, making it possible to turn over the engine by hand in starting. The exhaust-cam roller is shifted to engage the relief-cam in order to do this. It will be noticed from Fig. 84b that the valve-springs are held by sleeves which act as guides or crossheads for the valves.

The **make-and-break igniter** is operated by a push rod actuated by a crank on the end of the cam-shaft. A simple device is

used for advancing and retarding the spark. Current is supplied by a magneto driven by spur gears from the cam-shaft.

The **governor** is of the centrifugal-inertia type and is located in the flywheel. Its action is direct on a balanced throttle valve.

Lubrication of the crank-pin and cylinder is taken care of by sight-feed oil cups, one located on top of the cylinder, the other feeding the centrifugal oil ring on the crank-shaft which carries oil to the crank-pin. The main-bearings are oiled by chains running in reservoirs of oil. Other bearings are oiled by oil wells and grease cups.

MULTICYLINDER ENGINES

The **automobile** engine, or motor as it is usually termed, is one of the most highly perfected types of internal combustion engines. The development in this type has been comparatively recent. The four-cylinder motor did not come into general use in the United States until about 1903, and even after that date many manufacturers clung to the single- and two-cylinder motor for several years.

The peculiar demands made by the public have been largely responsible for the perfection of the automobile motor today. This type of motor is usually put in the hands of a man who knows nothing about machinery, as a whole, and whose knowledge of gas-engines is even less than his general knowledge of machinery. Therefore, the motor must run for long periods without any attention whatever. Lubrication must be automatic and unfailing for long periods. The cooling system must be simple and as unfailing as the oiling system. The ignition system, throughout, must be on a par with both the cooling and oiling systems. In addition, the general construction of the motor itself must be such as to withstand abuse such as no other piece of machinery is ever subjected to.

The Packard Motor.—In Fig. 85 is shown the exhaust side of the six-cylinder motor used in Packard motor cars. The **cylinders** are cast in blocks of three and are bolted to an aluminum base. The **exhaust-** and **inlet-valves** are of the poppet type and are placed on one side of the motor. The three electrical units shown in this view are, from left to right, the **starter,** the **magneto** and the **generator** which furnishes current for the storage battery used in connection with the starter and the lights. The

spiral gear drive of the generator and magneto is shown at the right.

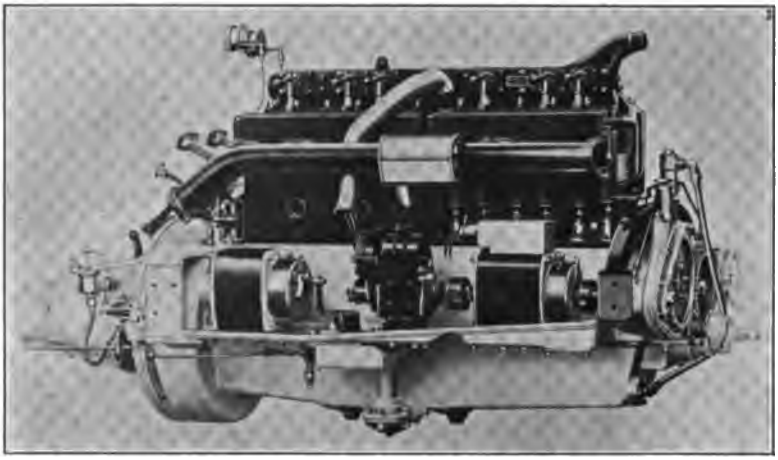


FIG. 85. — Six-cylinder Packard Motor — Exhaust Side.

In Fig. 86, the intake side, the **carbureter** is shown in the center of the motor, with the water-circulating **pump** at the left.

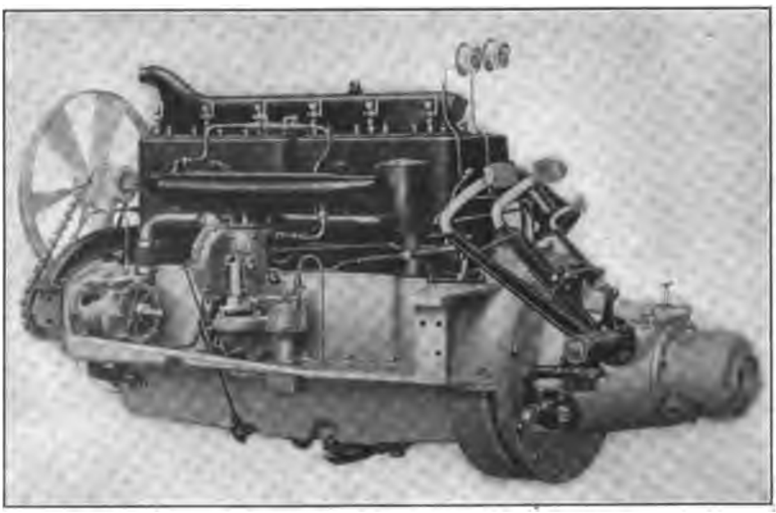


FIG. 86. — Six-cylinder Packard Motor — Intake Side.

The connection between the pump and the carbureter is the **hydraulic governor.** As the motor increases slightly in speed, the discharge pressure of the pump increases slightly, and this

increase in pressure moves a diaphragm which partially closes the throttle.

The hot-water connection to the top of the radiator is shown at the top and left of this view, but the connection between the bottom of the radiator and the circulating pump is not apparent.

In Fig. 87 is illustrated a partial section of this motor, showing the oiling system. The **oil pump** is shown at the right and above the gears, with its suction running to the low point in the crank case. From this pump oil is delivered to the cylinders, to the cam-shaft bearings and to the main-shaft bearings. The

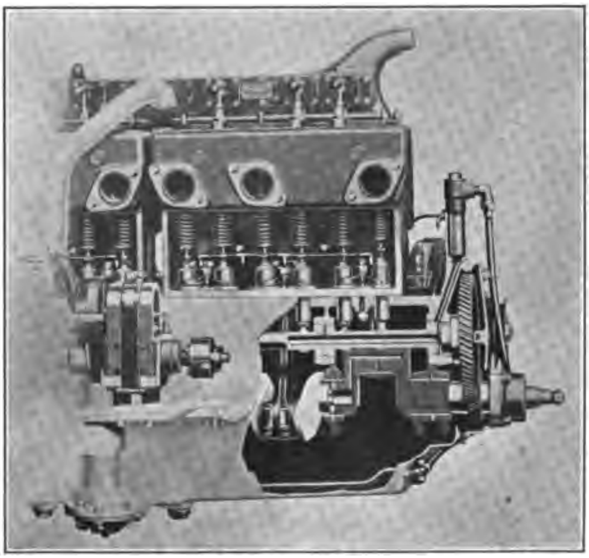
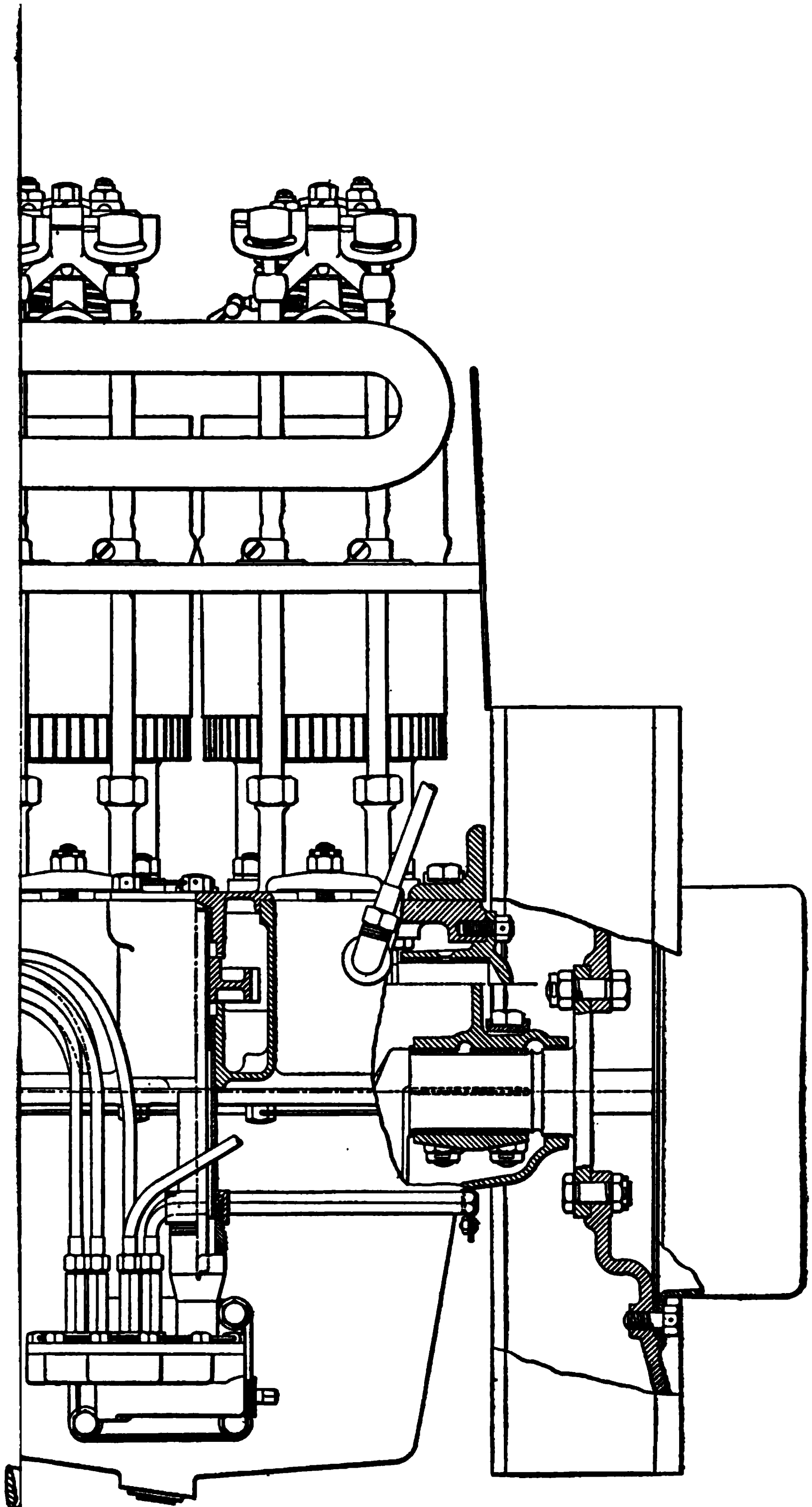


Fig. 87. — Six-cylinder Packard Motor — Partial Section.

main- and cam-shafts are hollow, and oil is delivered through holes in the shaft to each bearing. Oil also flows from the bearings into the hole in the main-shaft, then through holes in the crank-arms, to the crank-pin bearings.

The Franklin Motor. — The motor built by the H. H. Franklin Company, of Syracuse, N. Y., and used in their cars is shown in Fig. 88. The distinctive feature of this motor is the cooling, which is done by air. The cylinders are cast individually, as shown, with steel fins, parallel to the cylinder axis, cast into place. There are about fifty of these fins 8 inches long, $1\frac{3}{8}$ inch wide



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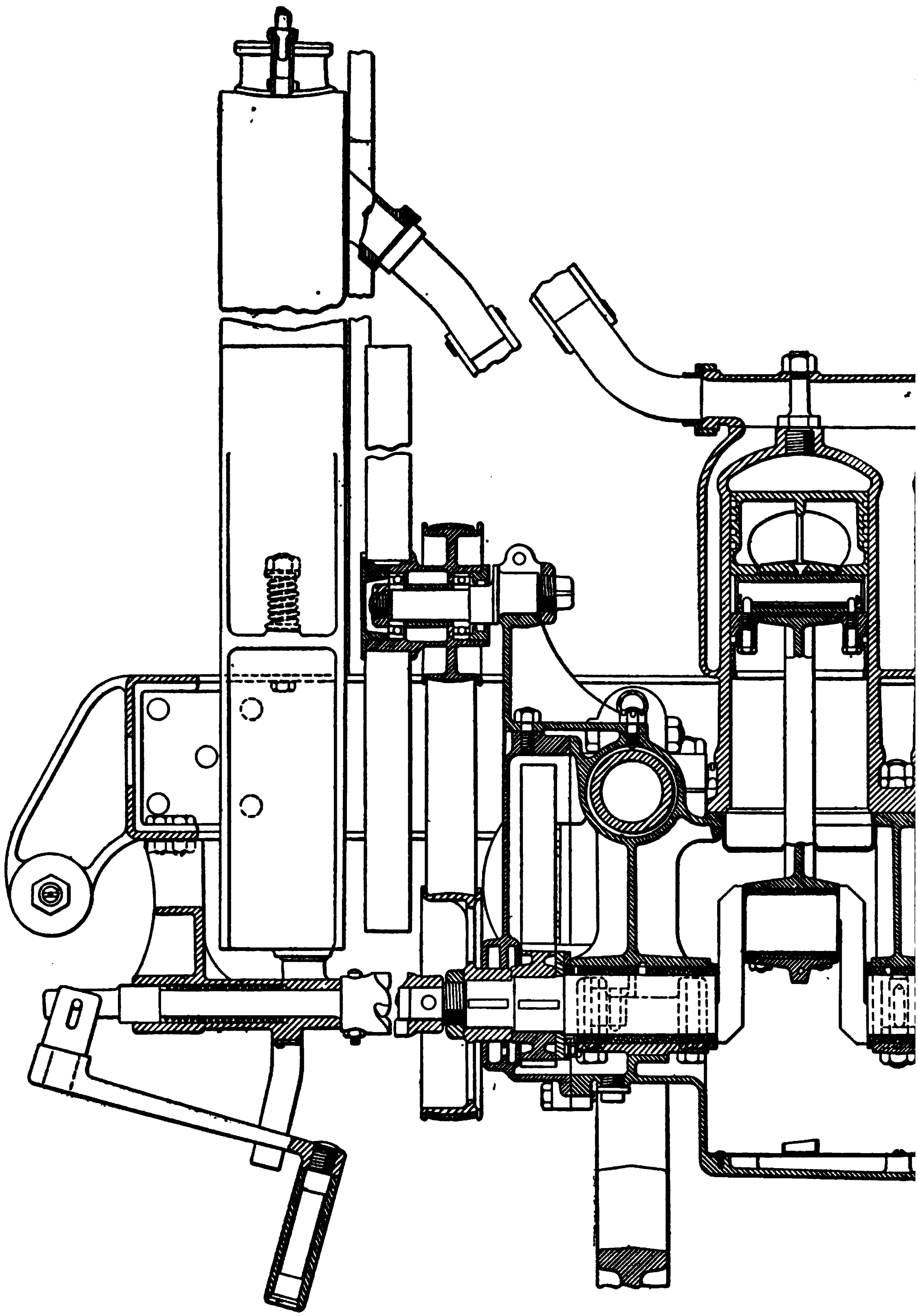
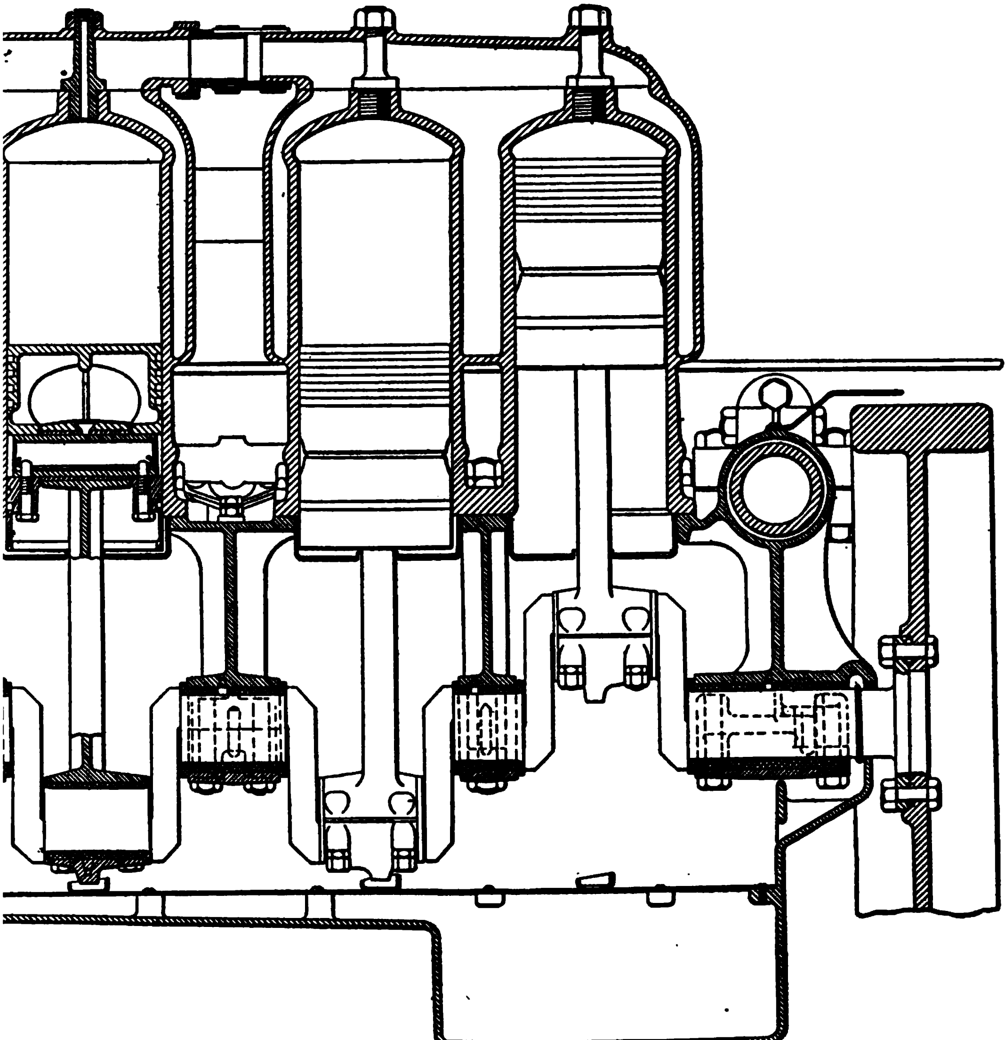


Fig. 89. — Locomobile Four-cylinder Trucl



Motor — Longitudinal Section. (Following Fig. 88.)

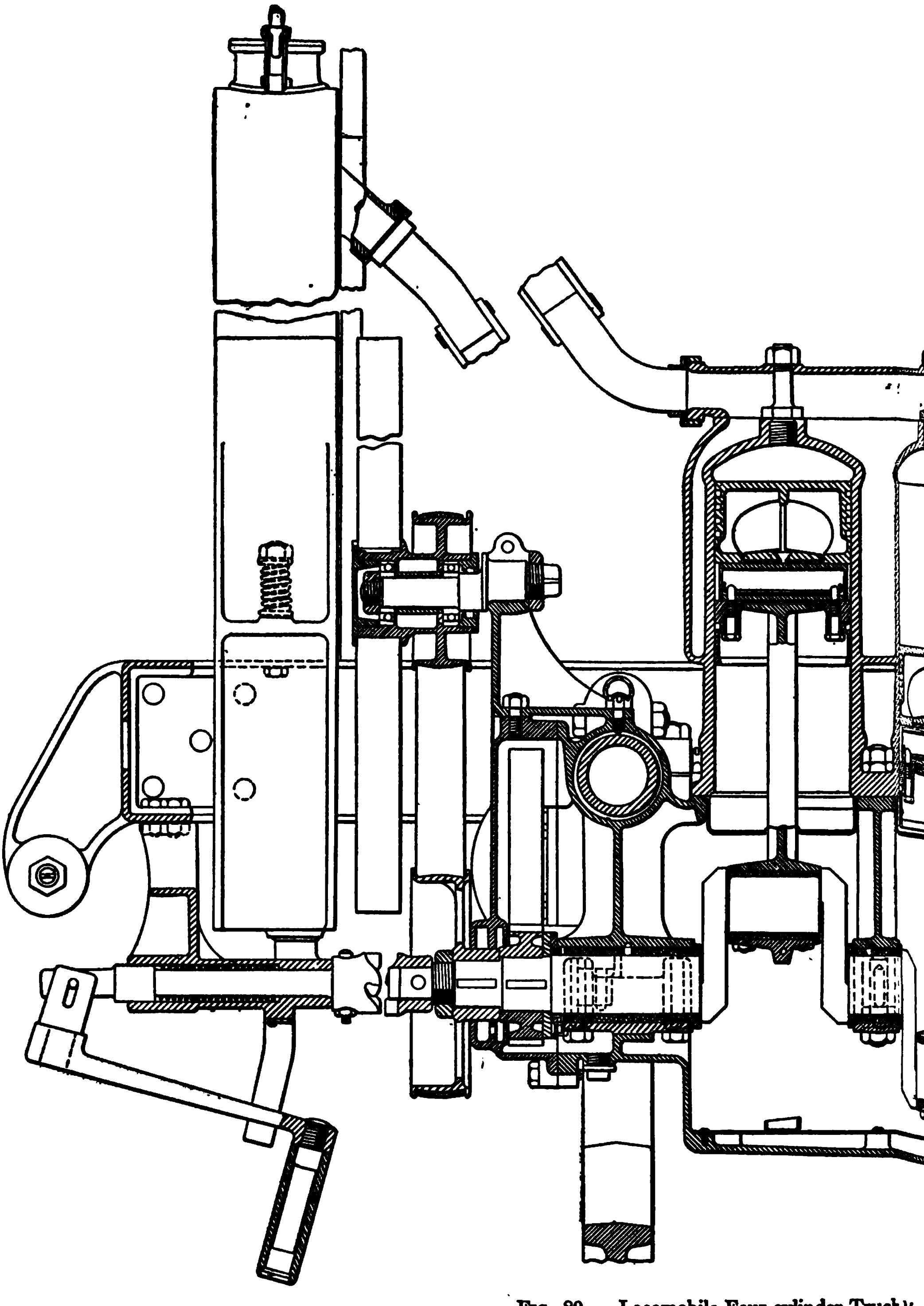
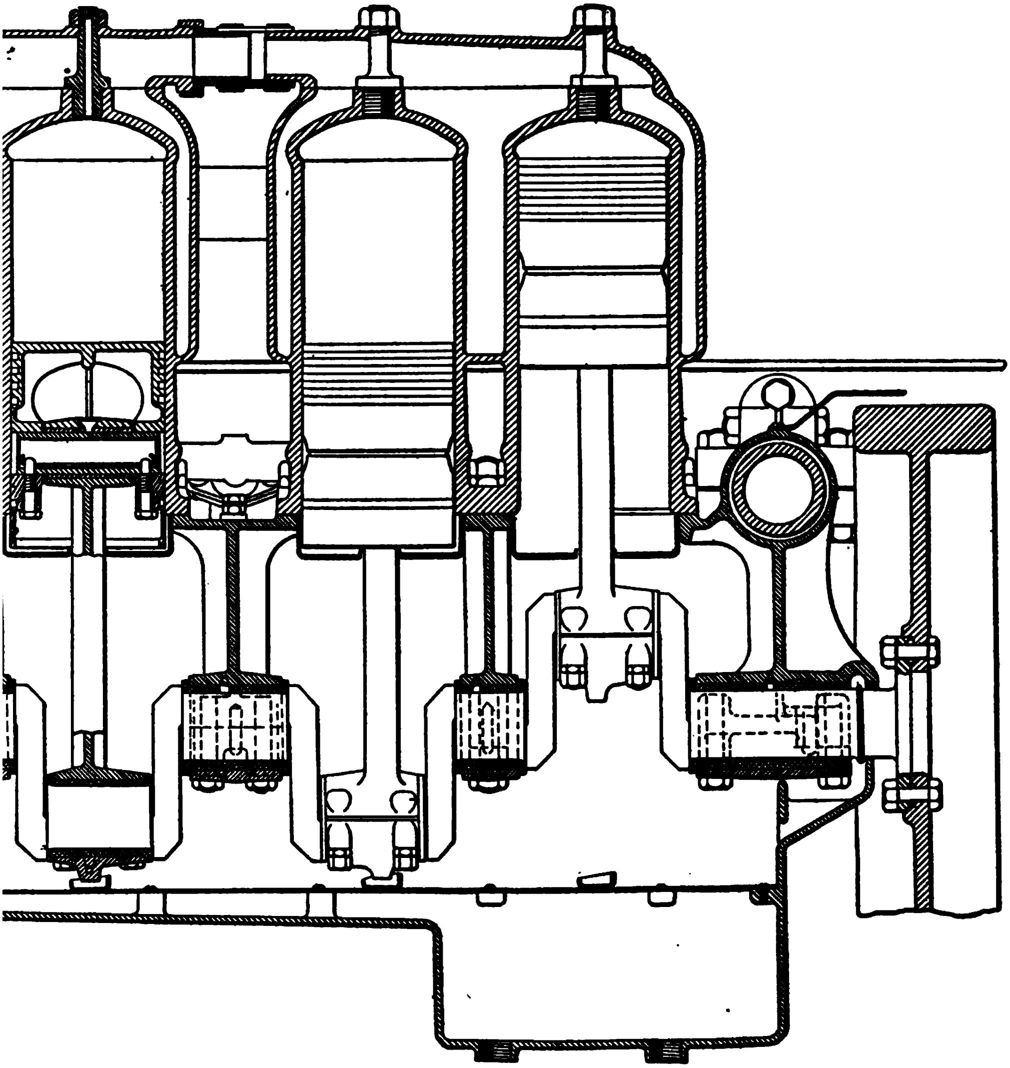


FIG. 89. — Locomobile Four-cylinder Truck Mot



: Motor—Longitudinal Section. (Following Fig. 88.)

cast in each cylinder, giving a cooling surface of 325 square inches in the fins alone.

Surrounding these fins are aluminum jackets, one for each cylinder. Air is drawn by the **fan vanes** on the flywheel down through these jackets, as the hood enclosing the engine fits around the base in such a way that the path inside the cylinder jackets and between the fins is the only one left for the air to take.

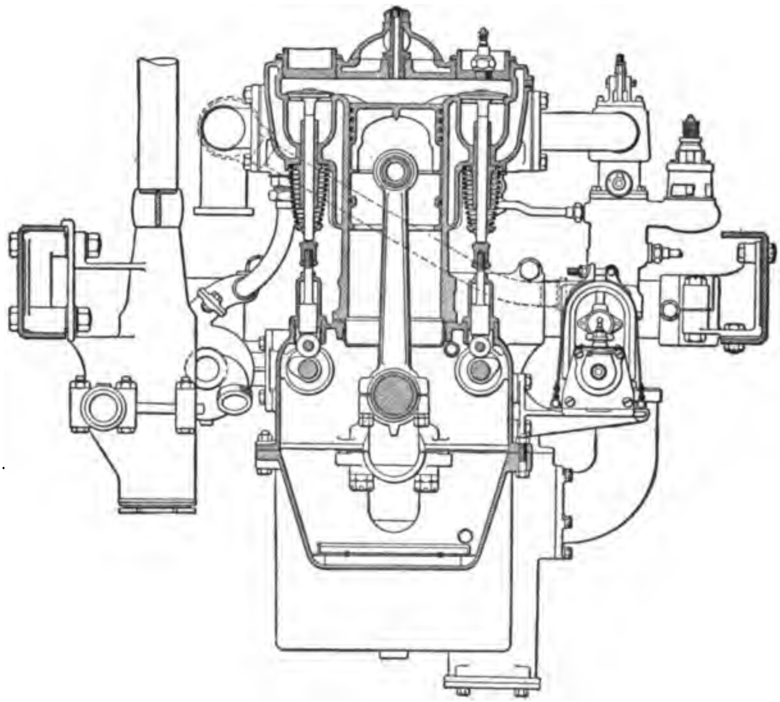


FIG. 90. — Locomobile Four-cylinder Truck Motor — Transverse Section.

The overhead **valves** are operated by vertical push rods and levers. The push rods are adjustable and the bearings of the levers have adjustments for taking up wear.

The **oiling system** is of the force-feed type, oil being delivered by a vertical rotary pump from the bottom of the crank-case as shown.

The Locomobile Motor. — The Locomobile four-cylinder truck motor is illustrated in Figs. 89 and 90. The diameter of the cylinder is 5 inches, stroke 6 inches, and the rating of the motor is 45 horse-power at 900 revolutions per minute.

The cylinders are cast in pairs with a jacket cast integral with them. The thickness of the cylinder tapers from $\frac{5}{16}$ inch at the bottom of the water jacket to $\frac{7}{16}$ inch at the bottom flange. The admission- and exhaust-ports are 2 inches in diameter, giving gas velocities of 5700 feet per minute with an engine speed of 900 revolutions per minute. Compression is 66 pounds gauge.

The **crank-case** is of government bronze. It is supported from the frame at front and rear by means of steel tubes, which rest and are held down in brackets, bolted to the frames by means of caps, bolts and nuts.

The **crank-shaft,** which is of alloy steel, heat treated, is supported from crank-case by means of caps, studs and nuts. The aggregate length of the five crank-shaft bearings is $18\frac{3}{8}$ inches.

The crank-shaft **gears** are steel drop forgings with spiral teeth, case hardened.

The **cam-shafts,** of steel, hardened and ground, also have five bearings. The cam-shaft **gears** are of cast-iron with spiral teeth.

The **valves** are steel drop forgings, $2\frac{1}{2}$ inches in diameter, with a conical seat of 45 degrees. The lifter guides are of phosphor bronze. The guides for the valves are separate iron castings forced into place and may be replaced when worn.

The **water pump** is of the centrifugal type and makes 1700 revolutions per minute with the motor running 900 revolutions.

The **oil pump** is driven by spiral gears from the exhaust cam-shaft. This pump is mounted and driven inside the crank-case. Oil is fed by pump into main supply pipe from which one pipe supplies oil to the oil-troughs and another pipe supplies it to crank-shaft bearings. Connecting-rods have scoops which dip into oil-troughs. Crank-pins are therefore oiled directly, and the wrist-pin, piston and cam-shaft bearings by splash.

Ignition is of the high-tension dual system, magneto and storage battery. The magneto is a Bosch, with coil for starting. The magneto has a fixed spark advance and the battery has a fixed spark at about dead center for starting.

The Stearns-Knight Motor. — There have been many attempts to utilize a valve for the gasoline automobile motor, that would not have of the disadvantages of the lift poppet valve. The earliest motors used rotating valves but these proved to be unsatisfactory and poppet valves took their place. Sliding

valves of different types have been proposed and some of them experimented with, but none have really proven successful until the Knight sleeve valve motor was brought out. This motor is illustrated in Figs. 91, 91a and 91b. Instead of the poppet valves, the valve action consists of two concentric sleeves sliding up and down between the piston and cylinder walls. These **sleeves** open and close wide slots or ports opening directly into the combustion chamber through which the inlet and exhaust

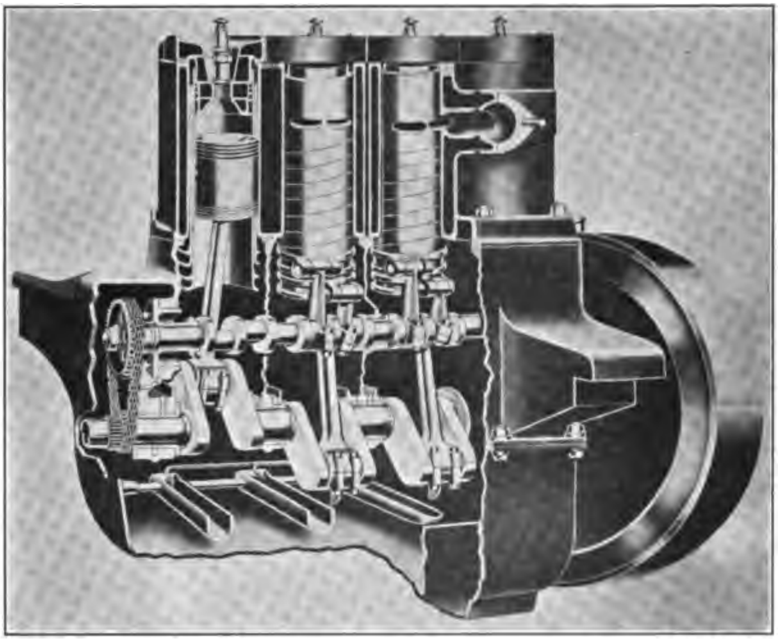


FIG. 91. — Stearns-Knight Four-cylinder Motor — Longitudinal Section.

gases pass. They are moved up and down by small connecting-rods from a crank- or eccentric-shaft in about the same manner as the slide-valves of a steam-engine are driven. This shaft is driven from the main crank-shaft by a silent chain with a 2 to 1 reduction.

The **eccentrics** which drive the two sleeves for one cylinder are set at about 80 degrees. The slots in the sleeves "register" at the proper time to give inlet and exhaust functions, and this period of registrations or overlapping of the ports is just when the velocity of the sleeve is a minimum. Thus, the events occur at

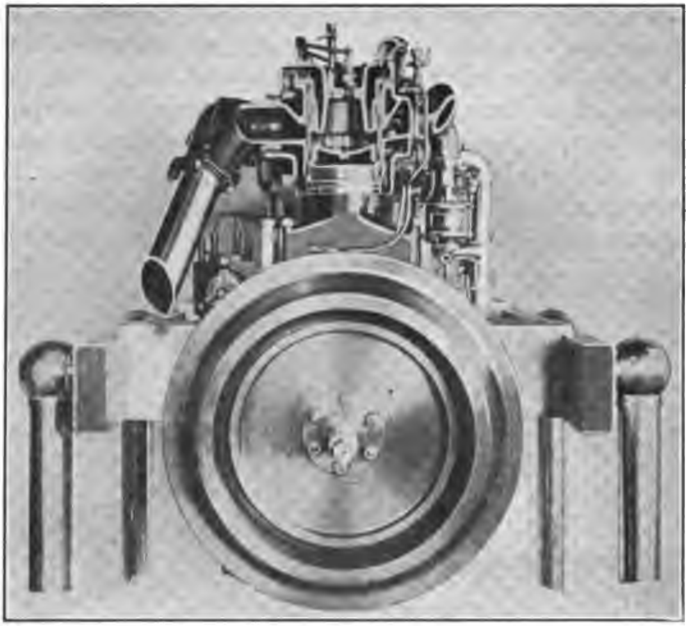


FIG. 91a. — Stearns-Knight Four-cylinder Motor — Transverse Section.

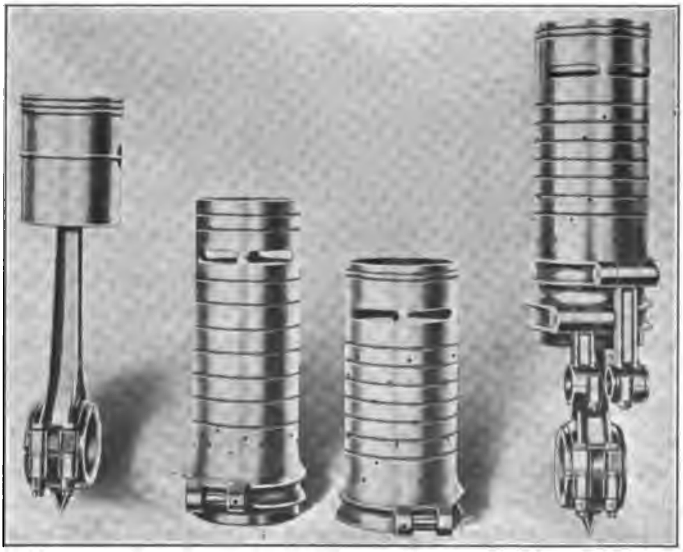


FIG. 91b. — Stearns-Knight Motor — Piston and Valve Sleeves.

the proper time, the ports open rapidly, remain open during the required period and then close sharply.

The **carbureter** used on the four-cylinder motor is of the double-jet type; one jet supplies gasoline for slow and intermediate speeds, and an auxiliary jet comes into service automatically at higher speeds.

The **oiling system** consists of an arrangement of movable troughs mounted in the bottom of the crank-case. The troughs are filled with oil at all times, the connecting-rods passing just over them. Scoops placed at the bottom of the rods dip into the oil. The lever actuating the movements of these troughs is connected to the throttle and as the throttle is opened wider and the load on the motor increased, the troughs are raised so that a greater supply of oil comes into use. The troughs are each fed from a separate pipe, which in turn is fed by a force feed pump. The pump is gear driven and with all the piping is contained in the bottom of the crank-case. In addition, an auxiliary oil supply is also connected to the throttle, but in such a manner that it becomes operative only when the motor is worked at excessively high speeds or when the load has become very heavy.

Ignition is by the dual system. It consists of an arrangement of the Mea high-tension magneto and a single vibrator coil, whereby a single set of spark-plugs affords a double system of ignition. The magneto is also chain driven.

MULTICYLINDER STATIONARY ENGINES

The multicylinder, vertical stationary engine has become very popular for small power plant work. Because it is multicylinder the torque on the shaft is more uniform than when a single cylinder is used. Also, having a number of cylinders, each cylinder is small for a given power compared to the single-cylinder type, and rotative speeds may be higher. Again, vertical engines require less floor space than horizontal engines, and they may be more easily balanced to avoid disagreeable vibration.

The Bruce-Macbeth Engine. — This engine is illustrated by a sectional drawing in Fig. 92. It will be noticed that the cylinder and jacket are cast together, with head cast separately. The **valves** are in the head, both inlet and exhaust being vertical. They are operated from cams on the cam-shaft which is shown

on the exhaust manifold. This **cam-shaft** is driven by means of a pair of spur gears and two pairs of bevels. On the vertical shaft is located the **governor.** The sleeve of the governor extends above the governor proper and is connected at its upper extremity to a rocker-arm which passes between the cylinders and is attached to the governor-valve illustrated in Fig. 53.

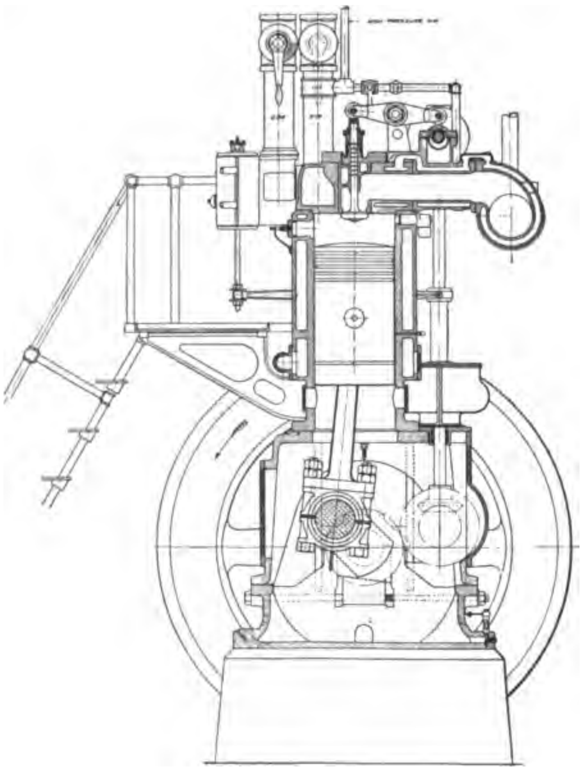


FIG. 92. — Bruce-Macbeth Multicylinder Vertical Gas-engine — Transverse Section.

The **crank-shaft bearings** are adjusted by a wedge on a long threaded bolt which passes out through the crank-case for convenience in making adjustments. The crank-shaft and crank- and wrist-pins are oiled by a splash system, the piston by a forced feed. The cam-shaft, it will be noticed, is oiled by ring oilers.

Westinghouse Engine. — The Westinghouse vertical three-cylinder engine is illustrated in Figs. 93 and 94, which show

longitudinal and transverse sections through the cylinders. The **cylinders** are cast separately, integral with their water jackets, and are bolted to a base. The **head** has a deep water jacket, and there is a water passage from the cylinder jacket to the head as shown in detail at the upper left-hand corner of Fig. 93.

The **connecting-rods** of this engine are rectangular in section with a marine end at the crank and a closed end at the wrist-pin. Adjustment at the crank-pin end is secured by the use of shims,

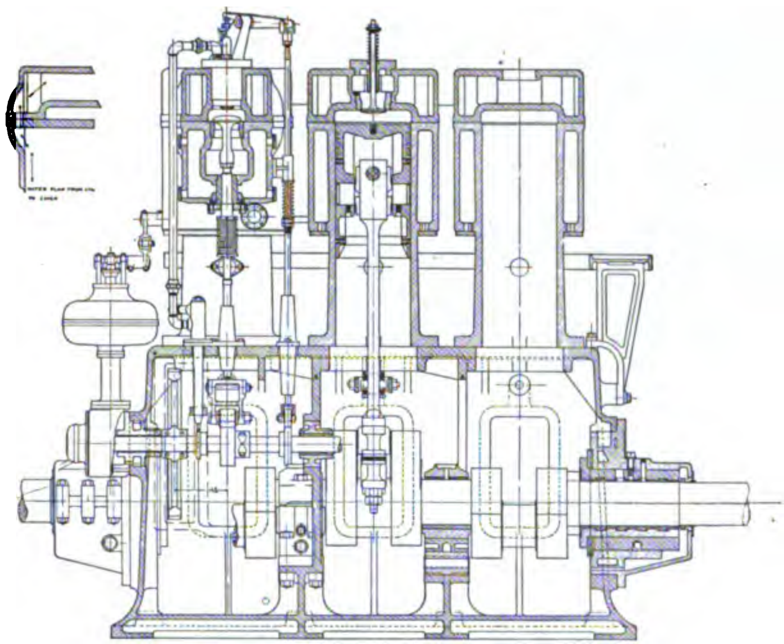


FIG. 93. — Westinghouse Three-cylinder Vertical Gas-engine — Longitudinal Section.

and at the wrist-pin end by means of a wedge. The **main-shaft bearings** are wedge adjusted in the vertical direction.

The **inlet-** and **exhaust-valves** are both vertical, the inlet opening downward in the head and the exhaust upward at one side. The **cam-shaft** is in the crank-case. The cam rollers are carried on levers as shown in Fig. 94. There is a rocker-arm at the top of the cylinder for each inlet-valve.

The **governor** is of the throttling type. The governor weights

operate a sleeve valve which throttles both gas and air, maintaining a constant mixture.

The Foos Engine.—This engine, illustrated in Fig. 95, is built in two-, three-, and four-cylinder units, from 35 to 325 horsepower. The valves of this engine are located in the head and each valve is driven from a separate cam on the **cam-shaft,** which is enclosed in the base. Push rods from the cam rollers operate rocker arms on the cylinder heads, which in turn open the valves.

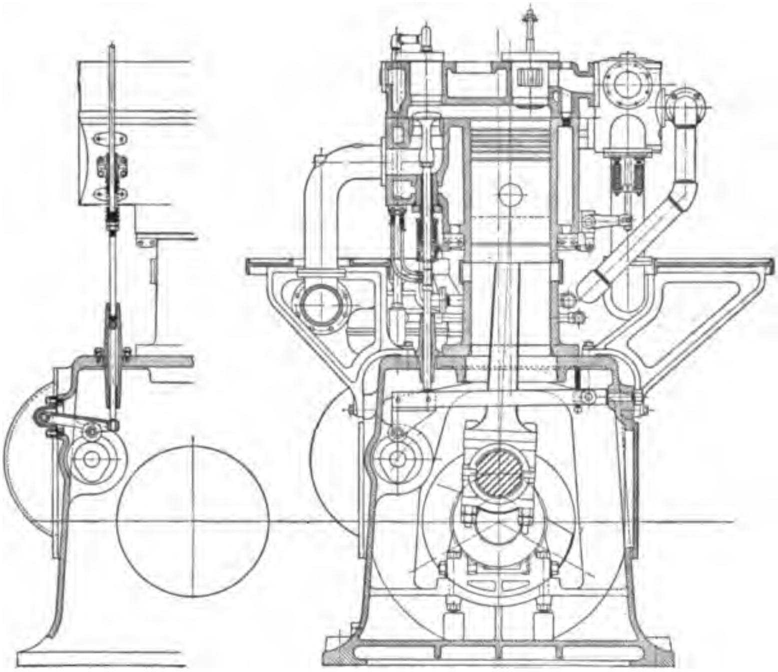


FIG. 94. — Transverse Section of Westinghouse Engine.

The cams are made in half to allow removal of any cam without disturbing the cam-shaft.

The **pistons** of this engine are equipped with six rings each, four at the top and two at the bottom to serve as oil rings. The rings are eccentric and cut with square lap joints.

Ignition is of the make-and-break type with igniters in the front of the cylinders. The time of ignition can be adjusted while the engine is running, each igniter being adjusted independently or all in unison.

The **connecting-rods** are of open-hearth forged steel. The de-

sign of the rod. is such that the piston-pin bearings can be adjusted by keys placed in line with the large openings in the side of the base. The crank-shaft is offset from the centerline of the cylinders so that the effect of angularity of the connecting-rod is decreased on the working strokes.

The **main-bearings** are cast separate from the base and turned to fit the corresponding bore of the base. Each bearing is removable without disturbing the shaft. Adjustment for wear is provided by liners under cap.

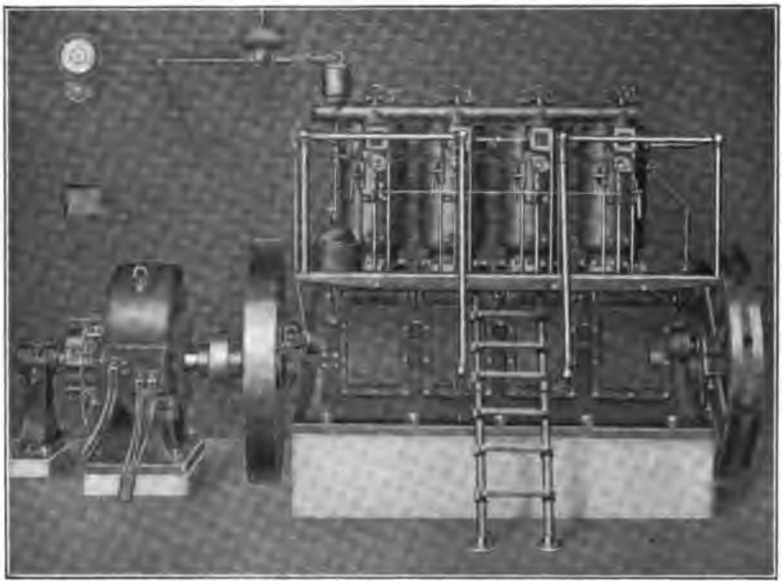


FIG. 95. — Foos Four-cylinder Vertical Gas-engine.

The governor proper is of the centrifugal type and controls balanced throttle valves located at each cylinder. The gears which drive the governor are placed inside the engine base.

The **lubrication** of the main-bearings, connecting-rod bearings, cam-shaft bearings, cylinders, pistons, cams and gears is accomplished by the splash system. Pockets and funnels assist in catching oil and conveying it to the crank- and wrist-pins and igniter gears.

The engine is started by compressed air working on every downward stroke of one cylinder until the other cylinders take up their regular operation.

Fairbanks-Morse Engine. — The Fairbanks-Morse four-cycle vertical engine for operating on kerosene and distillate running from 42° Baumé to 38 degrees is shown in Figs. 96 and 97. These engines are built in several sizes of two- and three-cylinder units from 50 to 200 horse-power.

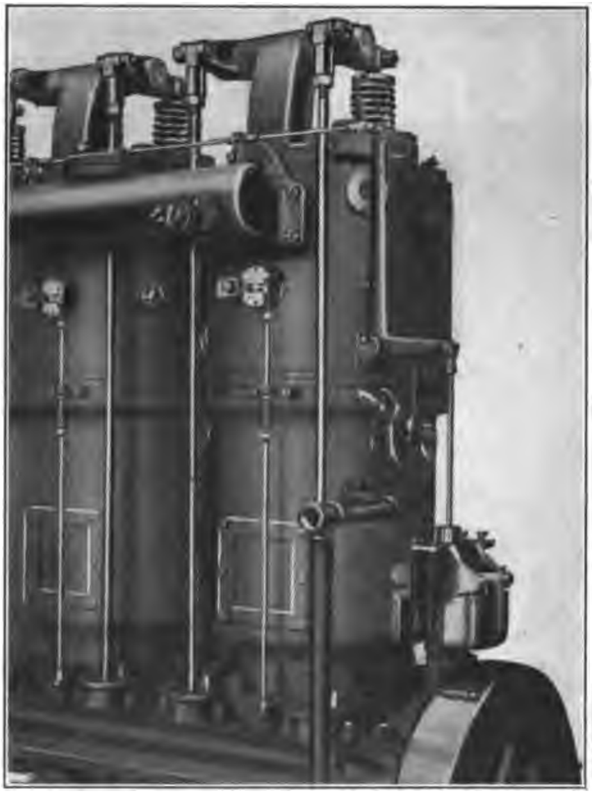


FIG. 96. — Valve-gear and Governor. Fairbanks-Morse Vertical Kerosene Engine.

The valves are placed in the heads, side by side, and are operated from cams through vertical push rods and rocker arms. The cam-shaft is placed inside the base as shown in Fig. 97.

On these engines the liquid fuel is fed in a finely-divided state directly into the air inlet passage at each cylinder head. Provision is also made for feeding a spray of water with the oil at each cylinder head.

The fly-ball **governor,** shown in Fig. 96, regulates the quantity of mixture fed to the cylinders.

One cylinder of each engine is fitted with an automatic **air starting gear.** The movement of the lever shown in Fig. 97 throws the inlet-valve out of action and slides into action two cams — one causing the exhaust-valve to open at every revolution and the other operating the compressed-air valve in such a

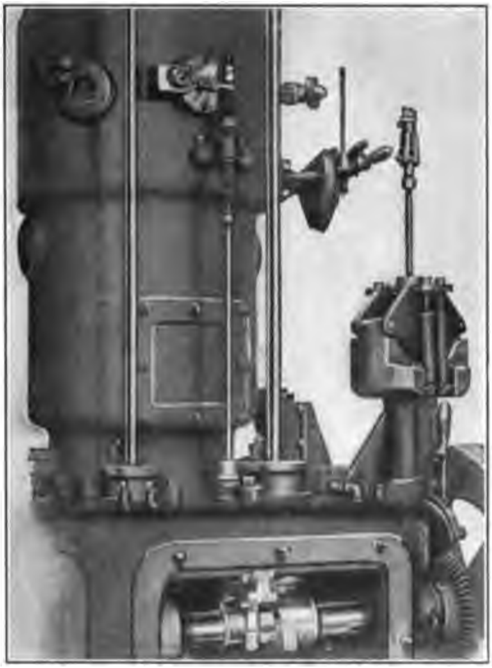


Fig. 97. — Cam-shaft and Igniter Details. Fairbanks-Morse Kerosene Engine.

manner that this one cylinder at the time being is converted into a single-acting air engine.

Ignition is by the make-and-break system. The point of ignition is easily changed by a small lever which acts on all cylinders simultaneously. This may be done either when the engine is running or when it is at rest. There is also an independent adjustment for each cylinder.

Lubrication for all important bearings is from a reservoir of oil located above the cylinder heads.

One of these engines, a three-cylinder, 80-horse-power unit is shown in Fig. 98. This engine operates on gasoline and is located in a private garage. It furnishes light and power for the house and grounds of a country estate.

The Rathbun Engine. — This engine, built by The Rathbun-Jones Engineering Company, of Toledo, Ohio, is illustrated in Fig. 67, Chapter XII.

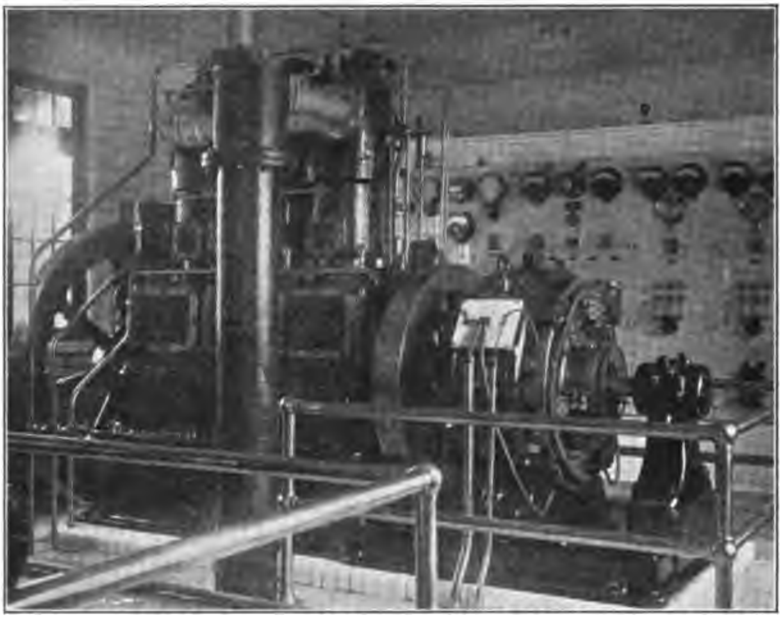


Fig. 98. — Fairbanks-Morse Three-cylinder Engine. Direct-connected to a Direct-current Generator.

It is of the four-cycle type, built in two-, three- and four-cylinder units up to 500 horse-power for the latter. The **valves** are placed in the cylinder head, which is bolted to the cylinder proper, the joint being ground to avoid the use of packing.

The **piston** is provided with an air pocket on the head to prevent the accumulation of oil on the hot center of the casting. Five **rings** are provided, the top being an ordinary snap ring, the others being of special construction whereby they always maintain pressure against the lower side of the slot instead of working from side to side with the motion of the piston. There

is also a connection through the cylinder wall which is always in communication with the space between the two lower rings and carries off the burned gases, dirty oil and products of wear from the cylinder instead of allowing them to work down into the engine housing. Pipes from these vents on the cylinder are carried to an oil-separating pot and from there a pipe to the open air disposes of the objectionable gases.

The **main-journal** boxes rest between jaws which form part of the base. The caps are lipped over the jaw and bolted down metal to metal. The journal boxes rest on wedges, adjustable from outside the engine. Means are provided for tramming the crank-shaft its entire length from the top of the housing which is planed parallel to the center of the shaft.

The **connecting-rod** is of forged steel with strap ends, adjustment at both crank and piston ends being obtained by wedge and bolt. Boxes at piston end are of phosphor bronze working on a hardened and ground steel pin. The boxes at the crank end are of malleable iron which are tinned and then lined with babbitt.

The **governor** is of the fly-ball type, operating directly on the throttle valve. The governor also changes the timing of the ignition to suit the compression as affected by the throttling. The governing is affected by the action of a two-disc balance valve operating directly on the mixture. The quality of the mixture is regulated by an additional valve in the gas line, and by a device peculiar to this engine, two settings of the mixture may be made, one for light load and another for full load.

It will be noticed that the **valves** of this engine are driven by **eccentrics** through the vertical push rods and levers. The eccentrics are placed in the crank-case.

The Nash Vertical Engine. — The engine shown in Figs. 99 and 100 is built by the National Meter Company of New York. This company puts out vertical units of one, two, three and four cylinders, from 6 to 425 horse-power. They are of the four-cycle type.

The frame and base are in two parts, split at the center line of the shaft. The **bearings,** of which there are five, are split and any sleeve, upper or lower, can be removed without disturbing the shaft.

The **valves,** inlet and exhaust, are side by side and open upward. They are of the same size and interchangeable. The stems

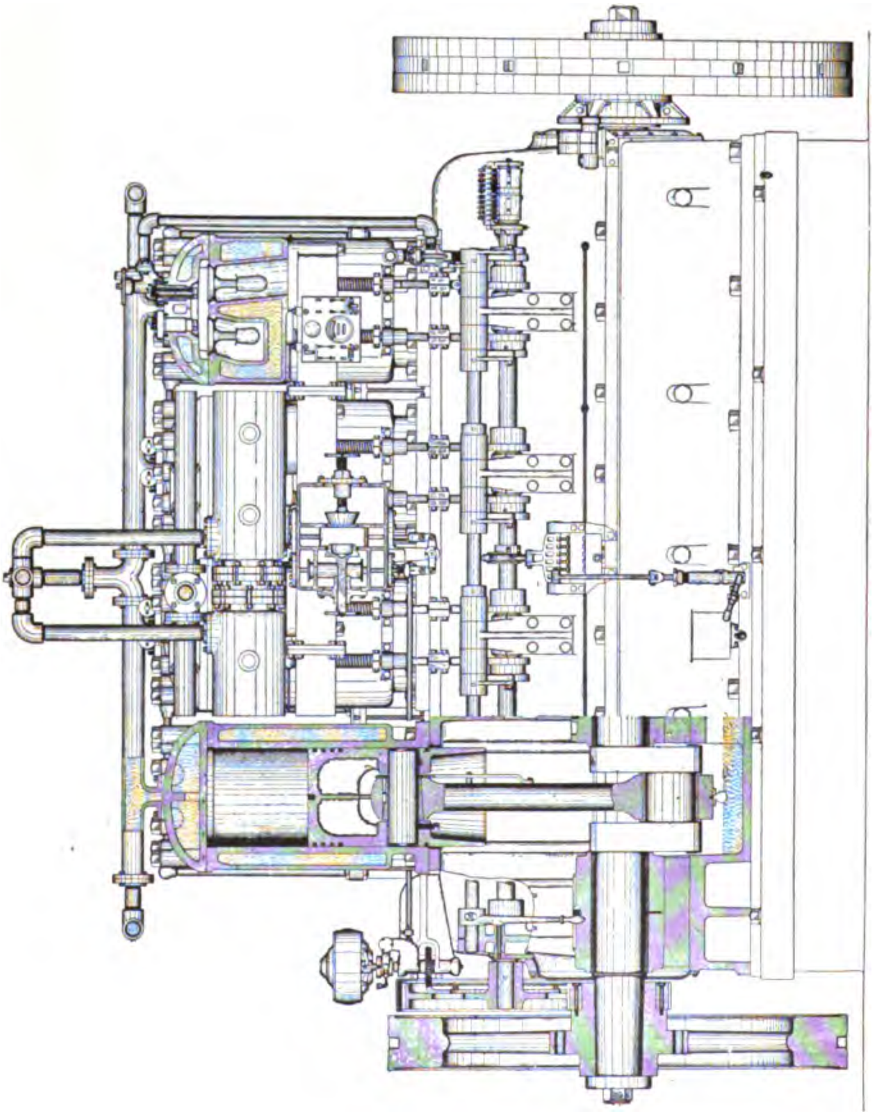


FIG. 99. — Four-cylinder Vertical Nash Gas-engine. National Meter Company.

are supported by two bearings or guides, with the spring between. The cam-shaft is placed outside of the engine base.

The **connecting-rod** is of the marine type at the crank-pin end but is solid at the wrist-pin end. Surrounding the wrist-pin is a bearing sleeve, split on the center line and tapered in the rod. This split cone is threaded at the end and carries a special thin nut for adjustment of wear. The wrench for making this adjustment is held in place on the rod with two cap screws.

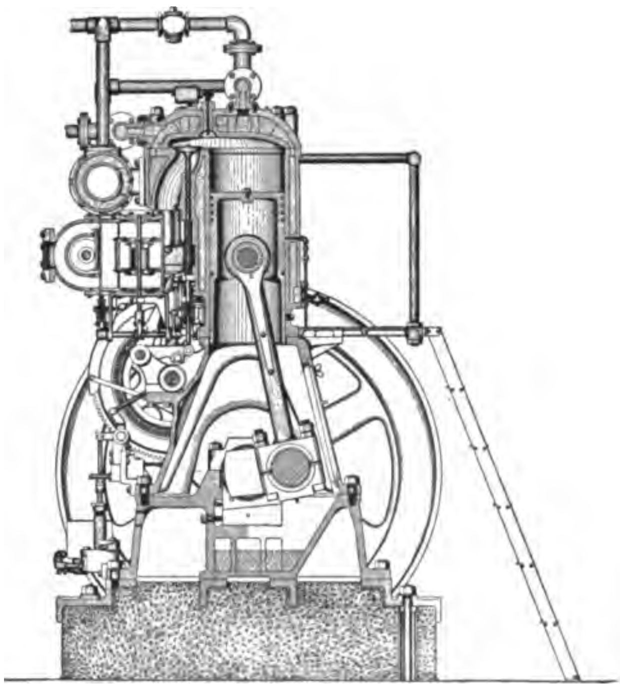


FIG. 100. — Traverse Section of Nash Engine.

The **governor** is located at the extreme left of Fig. 99. From the governor a rod extends to the vertical center line of the engine where it is connected to a balanced piston throttle valve, shown in Fig. 100. The valve serves to mix the gas and air, as well as to throttle the mixture.

The **ignition,** in the larger engines, consists of a magnetically operated make-and-break system. A solenoid coil, enclosed in an iron cylinder, upon excitation, actuates a cone, or bar, which

in turn operates the moving electrode, breaking the current and causing the spark. Electricity is supplied from storage batteries or from a special igniter-generator. The **timer** is adjusted to retard or advance the spark on all cylinders simultaneously.

The Sturtevant Gasoline-engine. — During the past five years a demand has grown for an engine or motor, similar to the automobile motor, for use in garages and isolated country homes. The engine of this type built by the B. F. Sturtevant Company of Boston is illustrated in Fig. 101. The design shows the influence of the automobile motor, and the characteristics which the two have in common only help to make the stationary unit more popular. The automobile motor is reliable and is understood by a large number of people who know nothing of any other type of internal combustion engine. For that reason, a stationary engine, which is to be used largely by men who own and are familiar with automobiles, should conform to the accepted general outlines of the automobile motor.

The Sturtevant Company manufactures this engine in three sizes, 5, 10 and 15 kilowatts capacity. The smallest size has 4 cylinders cast en bloc, $3\frac{1}{2}$ inches bore, 5 inches stroke. The 10-kilowatt engine has 4 and the 15-kilowatt 6 cylinders, 4 by 6 inches, cast in pairs. The speed of the smallest machine is 900 and of the two larger, 750 revolutions per minute.

The **cylinders** are of cast iron, the 5-kilowatt unit having the valves all on one side while the two larger units have the inlet- and exhaust-valves on opposite sides. In all three sizes the water jackets are cast integral with the cylinders.

The **pistons** are of the same quality of cast-iron as is used in the cylinders, and are provided with four rings.

The connecting-rods are of I section and drop forged. The crank-pin ends are of the marine type and are lined with interchangeable die-cast bushings of Parsons' White Brass. The piston-pin ends are bushed with phosphor bronze.

The **engine bases** of the 10- and 15-kilowatt sets are made of two iron castings, split horizontally on the center line of the crank-shaft bearings. A single casting is used for the base of the 5-kilowatt set, and in this case the generator is attached directly to the engine base. Separate sub-bases holding engine and generator are provided in the two larger sizes. The crank-shaft is carried on two, three and four bearings in the 5-, 10-

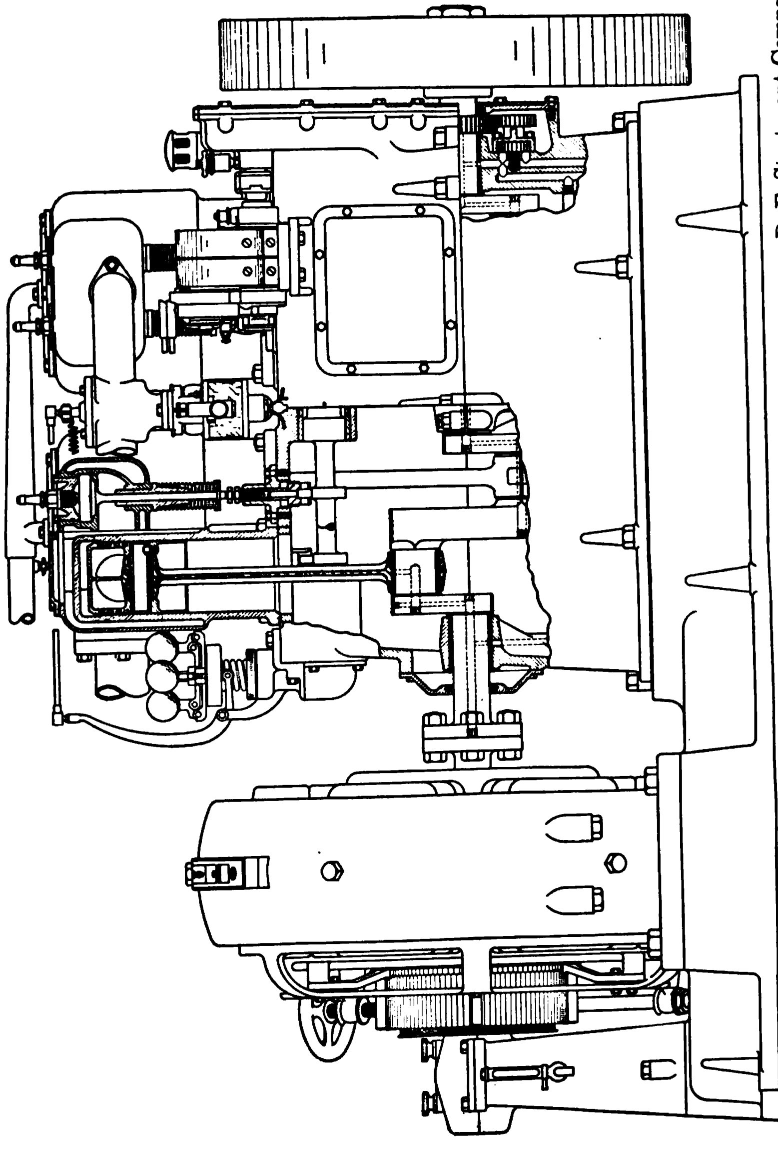


FIG. 101.—Four-cylinder Gasoline-engine. Direct-connected to Direct-current Generator. B. F. Sturtevant Company.

and 15-kilowatt sizes, respectively. These are lined with Parsons' White Brass interchangeable die-cast bearings.

Lubrication is accomplished by means of a forced-feed system. A gear pump, located in the base of the engine, furnishes oil under 20 pounds pressure to all bearings. The oil enters the main bearings and flows through the crank-shaft, which is drilled for this purpose, to the crank-pins, whence it passes up the connecting-rods to the piston-pins. The oil thrown off from the crank-shaft by centrifugal action serves to lubricate the piston in the cylinder. Oil under pressure is also supplied to the cam-shaft and the governor bearings. The oil falls back into the base, where it passes through a filter and is used over again.

The **governor** is of the centrifugal type and operates by throttling the mixture. Regulation is close enough so that the current may be used directly in lighting without the use of storage batteries to maintain constant voltage.

Cooling is accomplished by means of a circulating pump driven from one of the cam-shafts. A tank is furnished so that water may be pumped around through a closed system, the tank acting as a reservoir and a cooler.

Ignition is by means of a Bosch high-tension magneto driven from the cam-shaft. No batteries are used, the magneto being sufficient for starting.

CHAPTER XVI

CHARACTERISTIC OIL-ENGINES

Low-pressure Engines — Semi-Diesel Engines — Diesel Engines

General. — An oil-engine may be defined as an engine that uses oil for fuel, the oil being sprayed into the cylinder, or combustion chamber, during or at the end of the compression stroke. Some manufacturers give the name of "oil-engine" to engines that use carbureted fuel heavier than gasoline. This is misleading. There is practically no difference between an engine of this kind and a gas-engine. In both types fuel is drawn into the cylinder on the suction stroke and compressed on the return stroke. In the oil-engine, however, air alone is drawn into the cylinder on the suction stroke, and fuel oil is sprayed into the combustion chamber at the end of ordinary compression.

The oil-engine is manufactured in three distinct types. They might be termed the low-pressure, the semi-Diesel and the Diesel. The low-pressure type is so named from the fact that the compression is low, usually only 60 pounds per square inch. In this type fuel is usually sprayed into the combustion chamber during the compression stroke. Ignition is almost always secured by means of the hot-bulb, similar to the hot-tube ignition explained in Chapter IX, except that while the hot-bulb in the oil-engine is heated by outside means before starting the engine; for the rest, the bulb retains its heat from explosion to explosion. The pressures resulting from the explosion are not high, hardly as high as are found in some gas-engines. Combustion is not so thorough as in the higher pressure engines, and efficiency is correspondingly lower.

The semi-Diesel engine is a combination of the low-pressure engine and the Diesel type. The differences that exist between the three types lie in the compression and manner of igniting the fuel. The semi-Diesel engine uses a much higher compression than does the low pressure, but it is not so high as the compression in the pure Diesel type. From 150 to 300 pounds per square inch may be taken as the compression pressures used in the semi-Diesel

engines. Ignition in this type of engine is usually secured as in the low-pressure engine, that is, by a hot-bulb. This bulb must be heated by external means for starting, but after it is once heated thoroughly it retains heat from one explosion to another.

The injection of the fuel in the semi-Diesel engine cannot occur during compression as in the low-pressure type on account of premature ignition. The temperature corresponding to 250 pounds compression is sufficient to ignite almost all fuel oils, but it is not high enough to insure perfect ignition at every stroke. However, on account of the occasional pre-ignitions, it is necessary in the semi-Diesel type to inject the fuel oil at the end of compression.

The first successful Diesel engine was completed by Dr. Rudolf Diesel at Augsburg, Germany, in 1897. Since that time the manufacture of that type of engine has been taken up in almost every European country, and, to a certain extent, in the United States. The first American Diesel engine was built at St. Louis. This engine was built from designs acquired by Mr. Adolphus Busch, who, at that time, held the American patents and manufacturing rights. The first company to manufacture the Diesel engine in this country, the one organized by Mr. Busch, was the Diesel Motor Company of America, but this was soon superseded by the American Diesel Engine Company. In February, 1911, the company was reorganized under the name of the Busch-Sulzer Bros.-Diesel Engine Company, which has a factory at St. Louis, Missouri. In addition to this original company many other companies in the United States have experimented with this type of engine and a few have begun its manufacture. These companies are now building it in both the horizontal and vertical types, single- and multiple-cylinder, two- and four-cycle.

LOW-PRESSURE ENGINES

The "Giant" Fuel Oil-engine. — This engine, illustrated in Figs. 102 and 102a, is built by the Chicago Pneumatic Tool Company. It is of the low-pressure, two-cycle type, built to run on any petroleum fuel from gasoline to crude oil.

This engine illustrates the tendency towards crossheads for single-acting engines, instead of depending on the trunk piston to take the angular thrust of the connecting-rod. This arrangement is also more convenient for the two-cycle engine as it makes it possible to put in a scavenging cylinder between the motor cylinder

and crank-case as shown in the section. Leakage from this scavenging cylinder is prevented by a stuffing box on the piston-rod. If the crank-case were used for compressing scavenging air, the main-journals would also have to be made air tight. The crosshead arrangement also prevents any lubricating oil from entering the cylinder with the scavenging air.

Cylinder. — The cylinder of the "Giant" oil-engine is water cooled, the jacket being cast integral with the cylinder. The head is also water-jacketed, and arranged so that water flows directly from the cylinder to the head jackets.

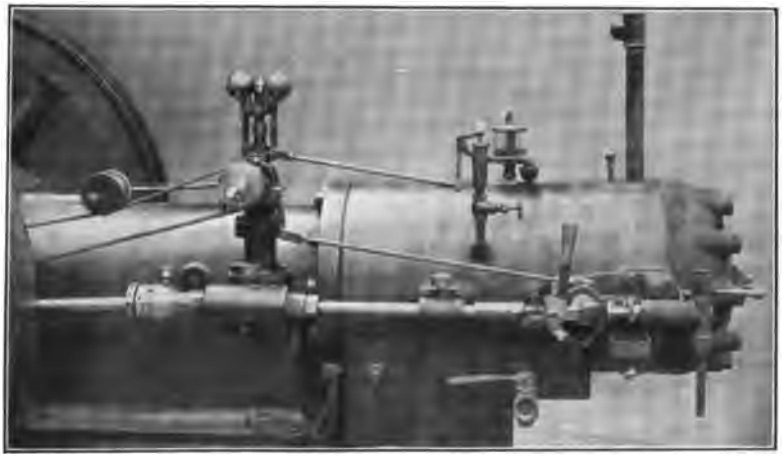


Fig. 102. — The "Giant" Fuel Oil-engine. Chicago Pneumatic Tool Company.

Piston. — The trunk type of piston is used, carrying four cast-iron eccentric rings. The rings are made wider than in four-cycle engines to insure smoothness in passing over the exhaust- and inlet-ports. The top half of the piston end is formed into a deflector to throw the scavenging air towards the head end of the cylinder.

Ignition. — A thin, circular plate is bolted to the piston, and after the engine is started and warmed up, this plate is hot enough to vaporize and ignite the fuel thrown against it. The **fuel oil pump** is a simple plunger pump with ball valves, two balls being used in series in both the inlet and discharge. The stroke of the plunger is varied by the governor, thus keeping the speed constant. The **fuel-injection-nozzle** is a combined

spray-nozzle and check-valve. It is screwed into the center of the cylinder head and directs the spray of fuel on to the plate bolted to the piston end.

Water Regulator. — Water is injected into the cylinder with the fuel and the quantity is varied to suit the load. The injection is through a needle valve which is controlled by the governor, the greater amount of water being admitted at heavy loads.

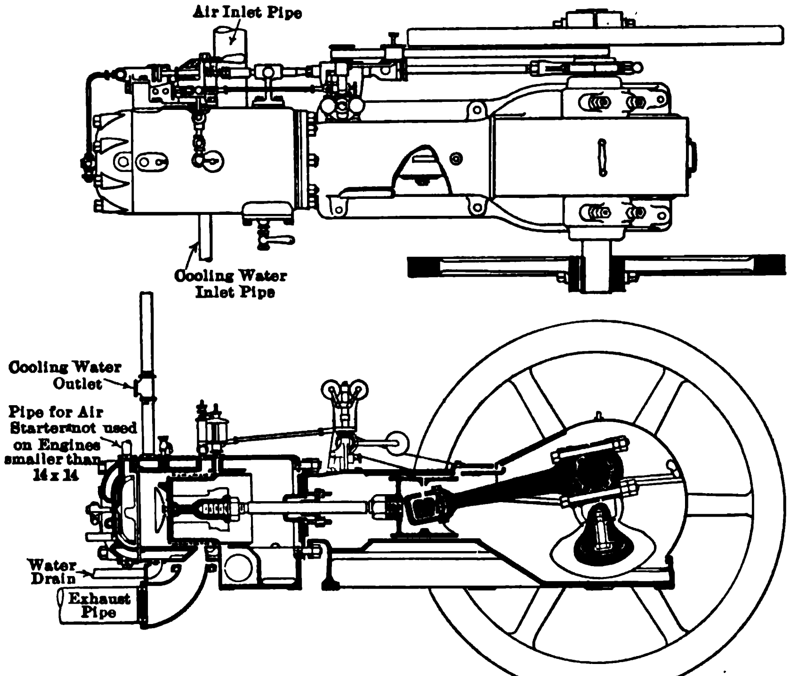


FIG. 102a. — Section and Plan of the "Giant" Fuel Oil-engine.

The Frame. — The engine frame forms the bored guides for the crosshead. The crank-case is entirely enclosed with an oil-tight cover. The **main-bearings** are cast integral with the frame, the caps being set at an angle of 45 degrees. Oil lips are cast on the outside of the bearings to catch and return to the crank-case any oil that works through the bearings.

The **shaft** is of the center-crank type, with bolted counter-weights. The **connecting-rod** is a steel forging. The wrist-pin end is of the solid type with a wedge adjustment and bronze

boxes. The crank-pin end is of the marine type, lined with babbitt metal.

Lubrication is by the splash system for all bearings except the cylinder, which is oiled by a sight-feed cup or a force-feed lubricator.

These engines are built in four sizes, 9 by 8, 10½ by 10, 12 by 12 and 14 by 14 inches, rated at 12, 18, 25 and 45 horse-power, respectively.

Mietz and Weiss Engine. — The two-cycle low-pressure oil-engine built by this company is made in various sizes and types. The stationary engine is manufactured in single- and two-cylinder

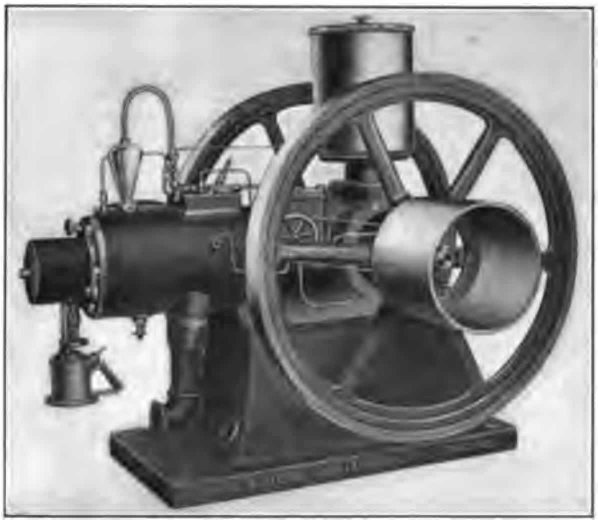


FIG. 103. — Mietz and Weiss Single-cylinder Oil-engine.

horizontal units, while the vertical type is made in one-, two-, three- and four-cylinder units. The horse-powers range from two to four hundred. The 2 horse-power unit runs at 600 and the 400 horse-power at 180 revolutions per minute.

Horizontal Type. — The horizontal type is illustrated in Figs. 103 and 104. The crank-case is used for compressing the charge of air which enters at the suction port shown above the base. As the piston travels on the outward, or working stroke, the exhaust port is uncovered first. After the pressure in the cylinder has dropped almost to that of the atmosphere, the inlet port on the top of the cylinder is uncovered and the air in the crank-case, which

is under a slight pressure, rushes in and cleans out the burned gases. Mixed with this air is the steam which is generated in the water jacket of the engine. The water jacket has no overflow, so that it is in reality a boiler generating steam at about atmospheric pressure, the steam thus generated being mixed with the charge.

Fuel Injection. — On the return stroke the air and steam are compressed in the combustion chamber to about 60 pounds gauge pressure. During this compression the fuel oil is injected

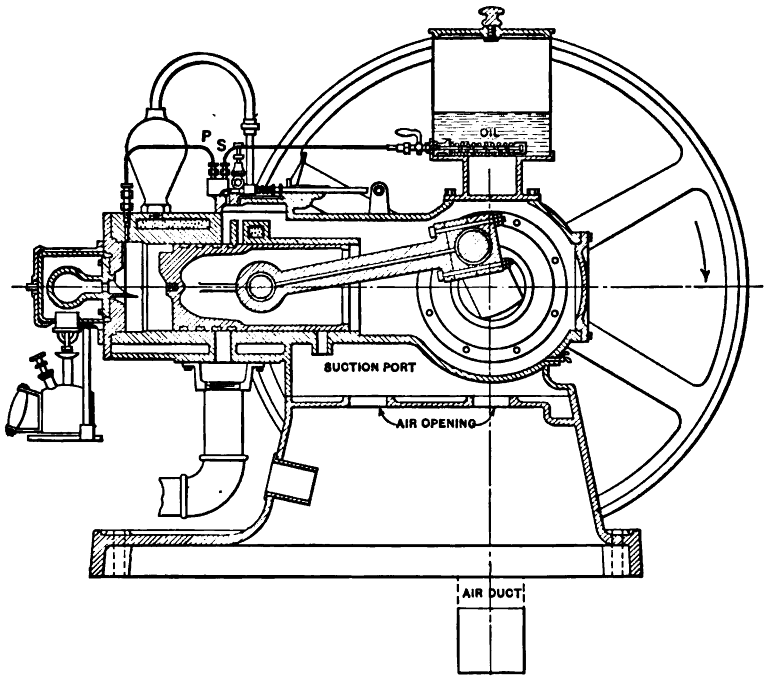


Fig. 104. — Longitudinal Section of Mietz and Weiss Engine.

by a small pump, and at the end of the compression stroke the mixture is forced into the hot-bulb at the end of the cylinder. No other means of igniting the charge is used.

Governing. — Located on the main-shaft of the horizontal units is an inertia governor which controls the eccentricity of the eccentric which operates the fuel pump, shown in Fig. 104. In the same figure is shown the rocker arm which is connected to the eccentric. The roller on this rocker arm taps the pump rod, thus forcing fuel oil into the cylinder. As the engine increases speed

the eccentricity of the pump eccentric is made less by the governor. This decreases the movement of the rocker and consequently shortens the stroke of the fuel pump.

The governor proper on the vertical type is of the centrifugal type, driven by gears and silent chain. Its action is the same as in the horizontal type.

Lubrication. — In the horizontal engines of 8 horse-power and over, and in the vertical of 4 horse-power and over, force-feed lubricators are used. These lubricators are located on the main-shaft and force oil to the cylinders, main-bearings and connecting-rods.

The Marine Type. — The marine type of engine is built much as the stationary vertical type, the units and powers being the same. The smaller sizes, up to 50 horse-power, are not made reversible, but the propeller is reversed by means of reversing friction clutches.

In the larger engines, from 75 to 400 horse-power, reversing is done by compressed air. The air is furnished by a small compressor driven by an eccentric on the main-shaft as shown in Fig. 105. For large units a separate compressor is also used so that air may be compressed before the main engine is started.

Reversing is easily accomplished as the engine has no valves. The operating lever being at the "stop" position, fuel oil and starting air are both shut off. The movement of the lever in either direction admits air to the proper cylinder to give the required direction. After the engine starts fuel oil is automatically fed to the cylinders and the normal operation takes place.

SEMI-DIESEL ENGINES

Fairbanks-Morse. — The Fairbanks-Morse small horizontal semi-Diesel engine is illustrated in Fig. 106. This two-cycle engine is made in four sizes from 10 to 25 horse-power. It operates on the usual crank-case compression plan except that air alone is used for scavenging and air alone is compressed on the compression stroke. The engine has exhaust and inlet ports which are uncovered by the piston so that there are no valves. Compression is carried to about 150 pounds and just before the end of the compression stroke, fuel oil is sprayed into the cylinder, and when it has vaporized due to the heat, combustion takes place owing to the hot-bulb shown at the end of the cylinder.

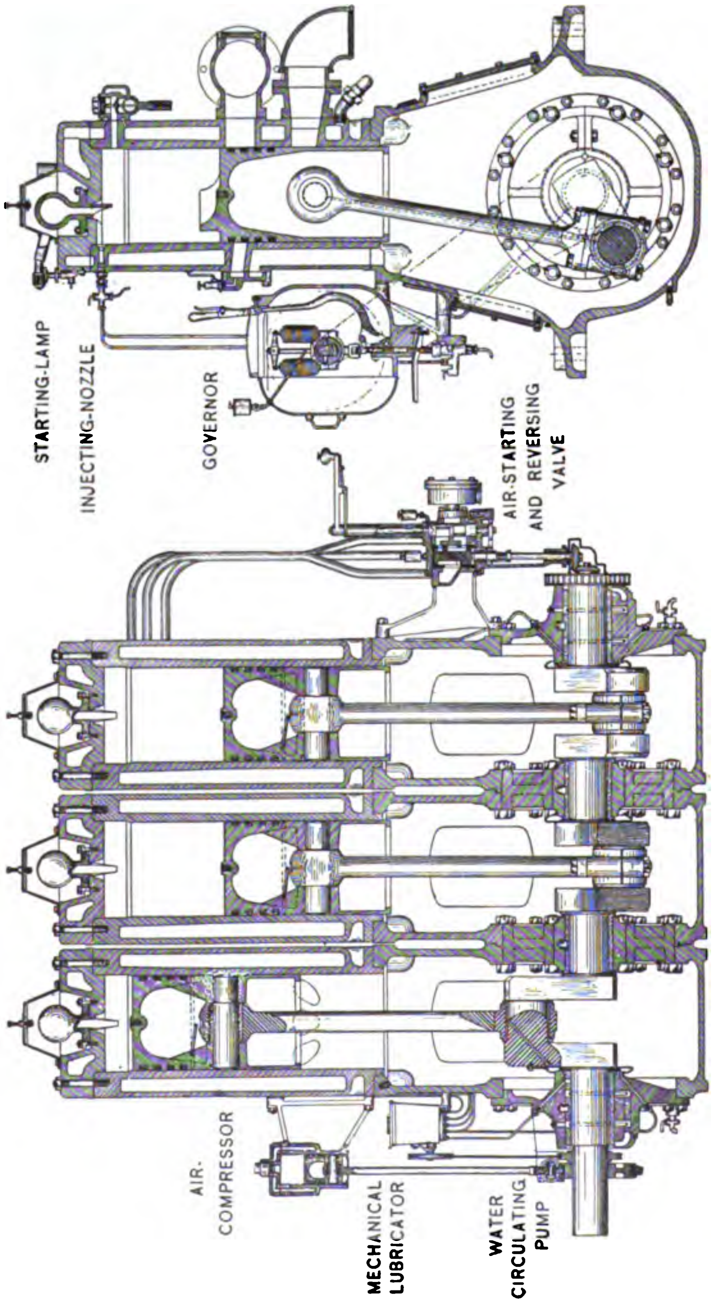


Fig. 105. — Mietz and Weiss Vertical Marine Engine.

Details of Construction. — The **cylinder,** which is water cooled, is cast separate from the frame and is bolted to it. The **shaft** is a solid forging and has cast-iron counterweights bolted to it. The **main-bearings** are turned and centered in a bored fit in the frame. The **bearing liners** are die-cast of special bearing metal and are made in halves. The upper halves of the liners and bearings may be removed without disturbing the lower halves.

The **piston** carries four narrow packing rings, three at the head end and the fourth at the other end. The **piston-pin** is made hollow and is prevented from turning by a key. The **connecting-rod** is of open-hearth steel, die forged. The piston end of the rod is a closed eye with a bronze bushing. The

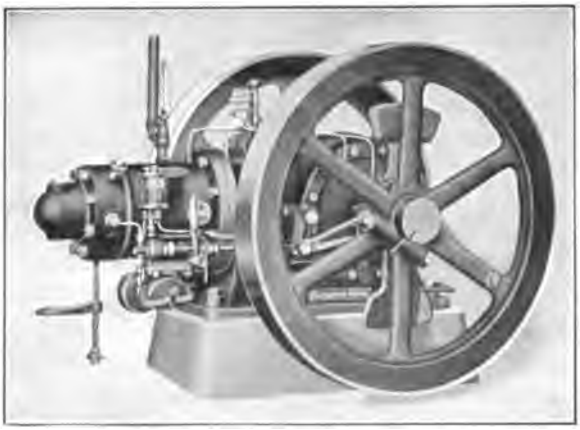


FIG. 106. — Fairbanks-Morse Two-cycle Semi-Diesel Engine.

crank-pin end is of the marine type with two vanadium steel bolts, the liners being of a special bearing metal. Adjustment is secured by the use of shims.

The **fuel injection pump** has a hardened steel plunger, with steel valves on bronze seats. The **governor** automatically alters the stroke of the pump to suit the load on the engine. The governor is of the flywheel type with large weights as shown. These weights are placed in such a way that they are not in balance but this is taken care of in the rim of the wheel. As mentioned before the weight of the crank and the reciprocating parts is balanced by the counterweights on the cranks.

A **lubricating oil reservoir** is mounted on the engine frame. A pipe connection from the top of the crank-case to the top of the

oil reservoir transmits air pressure to the top surface of the oil, thus forcing through a riser to the manifold which carries five sight-feeds, each separately adjustable. This pressure is held by a check valve in the air pipe. A three-way cock in this air pipe, when turned, opens a vent to the oil space for stopping the flow of oil when shutting down the engine or filling the reservoir.

One of the **sight-feeds** leads to the main cylinder, lubricating the piston and piston-pin. Two others lead to the main-bearings. The fourth leads to the crank-pin, the oil being carried directly to the bearing surface by a centrifugal oil ring and pipe and holes drilled in the crank-shaft itself.

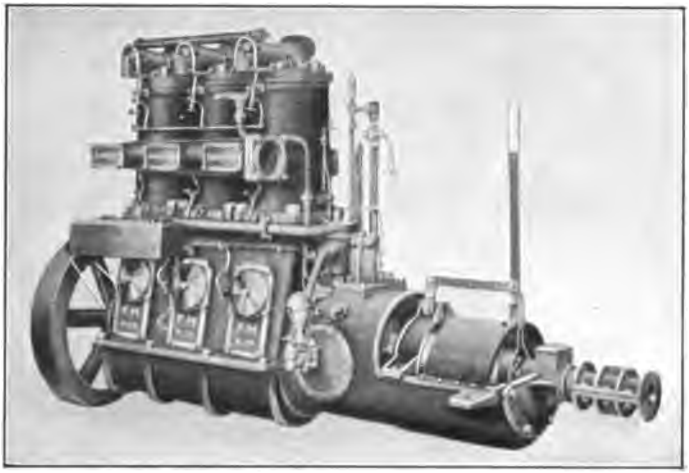


Fig. 106a. — Fairbanks-Morse Semi-Diesel Marine Engine.

The oil which drips from the crank-pin and piston is drained into the crank-case, which is provided with a drain pipe. The crank-shaft at the outer end of each main-bearing carries an oil thrower ring, and any surplus oil from the bearings is caught in the annular space at the outer end of each bearing and piped to the crank-case drain.

The Fairbanks-Morse Marine Engine. — This oil-engine, shown in Fig. 106a, is similar in general construction to the horizontal type already described. It is built in one-, two- and three-cylinder units, the diameter of each being 8 inches with a stroke of 10 inches. The rated horse-powers are 15, 30 and 45 respectively.

In the marine engine the crank end of the **connecting-rod** is

made up of two bronze shells, lined with babbitt metal, and bolted to the *T* end of the rod proper. The piston end is similar to that in the stationary engine. The **piston** carries six rings instead of four as in the stationary type. The engine is started with a compressed-air **starter** controlled by a hand throttle, which admits compressed air to a second valve, which is timed in relation to the motion of the piston in one of the cylinders. A water-cooled **compressor** is placed on the engine itself and is driven from an eccentric on the crank-shaft. This compressor is of sufficient capacity to furnish air for starting and for blowing the whistle.

The **fuel pump** is driven by a cam on the main-shaft. Each cylinder has a separate fuel pump avoiding any possible troubles in a distributing valve. The **governor** is of the fly-ball type and controls the speed of the engine by regulating the stroke of the pumps. The speed is also under the control of the operator by means of a hand lever, which controls the governor and permits a wide range of speed. The governor holds the engine at any speed set by the hand lever.

The engine is furnished with a mechanical force-feed **lubricator** driven by a belt from the crank-shaft. All the main-bearings of the engine, as well as the cylinder and piston, are lubricated by separate feeds from this positive feed oiler.

This engine is built to be reversed, this operation being done by a reverse gear.

De La Vergne Engine. — This engine, illustrated in Figs. 107, 108 and 109, is styled by the manufacturers as the Type F. H. It is a four-cycle engine and is operated on comparatively high compression — about 300 pounds per square inch.

The **cylinder** is cast in two parts, the jacket being integral with the frame, while the liner is separate. The liner is held in place at one end by the cylinder head, while at the other end there is a packed joint to allow for expansion.

The **piston** carries 7 rings and is conical on the end. This conical shape allows for expansion to a certain extent and it also allows a shape of combustion chamber which makes it possible to get in the valves and hot-tube.

The **connecting-rod** has a closed end with wedge adjustment at the piston end. The crank-pin end is of the marine type. The **shaft** is forged solid and the counterweights are held onto the cranks by tangent keys. The **cam-shaft** is driven from the

main-shaft through spiral gears. The governor, fuel-pump, inlet-, exhaust- and fuel-injection-valves are driven from the cam-shaft.

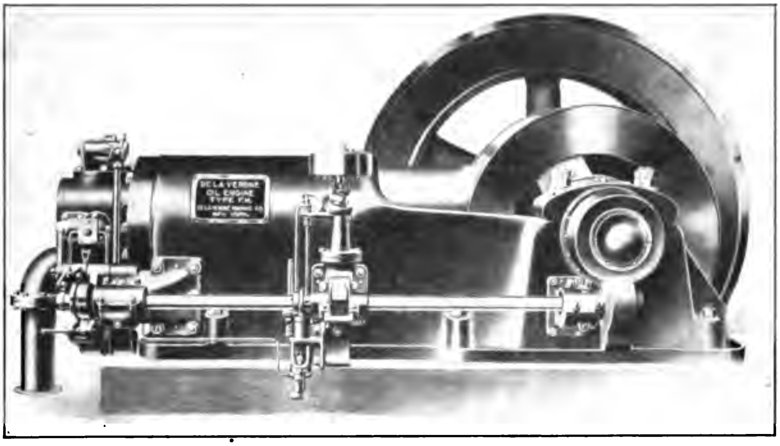


Fig. 107. — De La Vergne Type F. H. Oil-engine.

Fuel Injection Valve. — In this engine, since the compression is 300 pounds per square inch, it is necessary to use compressed air to spray the oil into the combustion chamber. The oil and

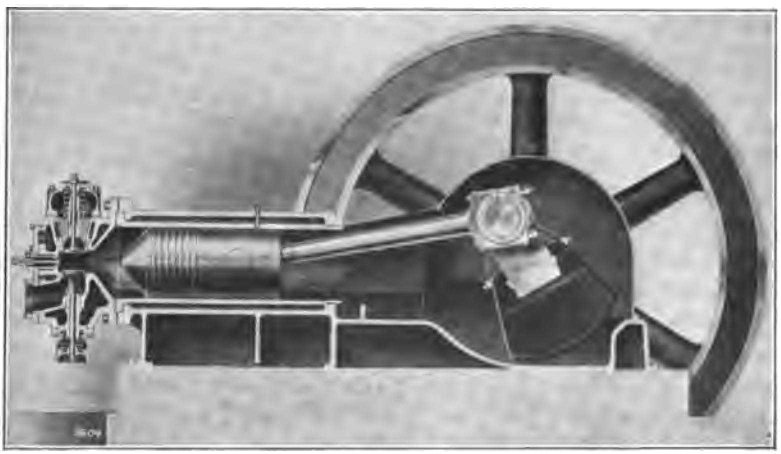


Fig. 108. Longitudinal Section De La Vergne F. H. Engine.

compressed air are admitted on opposite sides of a sleeve which is enclosed in a close-fitting outer envelope comprising the body of the valve. As the oil and air proceed side by side along the out-

side of the sleeve, they are forced to pass through a series of chambers or grooves separated by a system of fine diagonal channels on the sleeve. In this way a thorough mixture and subdivision of the oil is brought about.

The **needle valve** by which the charge is admitted to the cylinder is about one-half inch in diameter. It is held closed by a spring and is actuated by a cam on the cam-shaft. This spray valve is located on the side of the combustion chamber directly opposite the vaporizer, as shown in Fig. 109. Just before the piston reaches dead center on the compression stroke, the spray

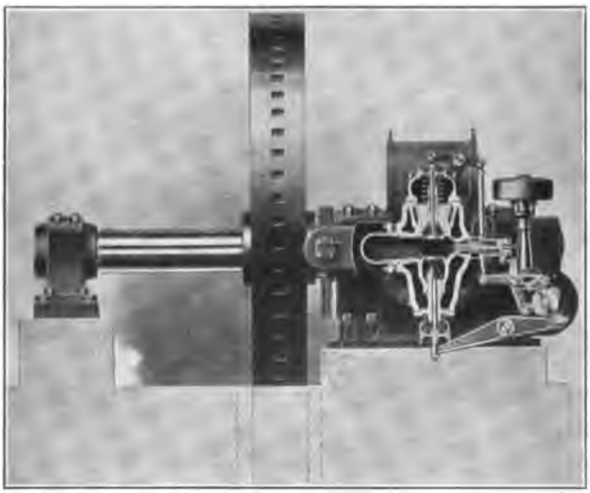


FIG. 109. — Section Through Combustion Chamber. De La Vergne F. H. Engine.

valve admits the finely-divided oil into the vaporizer where vaporization is completed and ignition occurs, due to the high temperature of the vaporizer. This vaporizer retains its heat from one explosion to another, but must be heated by a torch before the engine may be started cold.

The **air compressor** which furnishes high-pressure air used in the spray valve is a two-stage unit driven from the main-shaft by an eccentric. This compressor is between the cylinder and the flywheel. The end of the high-pressure cylinder is visible in Fig. 109. In the first stage air is compressed to 150 pounds and stored in a tank so as to be available for starting the engine. The second

stage of the compressor is quite small in capacity and handles only enough air to spray the oil from stroke to stroke.

The **governor** regulates the speed of the engine by altering the length of the stroke of the fuel pump. It accomplishes this by shifting the point of contact between the pump-lever and its actuating cam.

This engine is built with four sizes of cylinders, 14 by 24, 17 by 27½, 20 by 34½ and 23 by 40 inches. In single-cylinder units they are rated at 60, 100, 140 and 200 brake horse-power respectively. The same cylinders are also used in twin- and four-cylinder units with a corresponding increase in horse-power.

DIESEL-TYPE ENGINES

The Busch-Sulzer-Bros.-Diesel Engine. — The new engines of medium capacities built by this company and illustrated in Fig. 110 are of the four-cycle, single-acting, enclosed crank-case, four-cylinder, high-speed Diesel type. They are built in five sizes, *i.e.*, 120, 160, 240, 350 and 500 brake horse-power, running, respectively, at 300, 276, 257, 225 and 200 revolutions per minute. The general design of all sizes is kept substantially the same, deviating only where increasing size demands it. Considering one working cylinder with head, valves, piston, connecting-rod, etc., as a complete unit, then four such units are required to make up an engine.

The **base-plate** is a single casting containing the support for the crank-shaft bearings and the flanges for attaching the frame. The lower part serves as a collecting basin for the lubricating oil coming from the bearings. The parts intended to carry the bearings for the crank-shaft are of girder-like construction.

The crank-case, or frame, is also cast in one piece and forms the intermediate link between base-plate and cylinders. It is bolted all around to the top of the base-plate, thus helping to stiffen the latter. It is provided on both sides with doors for inspecting the running parts and the bearings inside. On the two larger sizes heavy tie rods between frame and base are provided to relieve the crank-case from the strains due to the cylinder forces.

The **cylinder** proper is again an independent casting, bolted to the crank-case and provided with a simple barrel-shaped **liner.** The latter is free to expand lengthwise, thus avoiding temperature strains and changes in the alignment. The liner is made of a hard close-grain special cast-iron pressed into the

so-called cylinder jacket, thus forming a water-jacket for cooling the cylinder walls exposed to the high temperatures of the gases of combustion. A registered joint at the top, between cylinder liner and head, prevents all leakage from inside the cylinder. A projecting bracket forming part of the cylinder jacket serves as a support for the cam-shaft casing.

The **cylinder head** casting contains all the valves necessary for the operation of the working cycle, *i.e.*, the fuel valve, starting-valve, suction- or intake-valve and the exhaust-valve. It is bolted

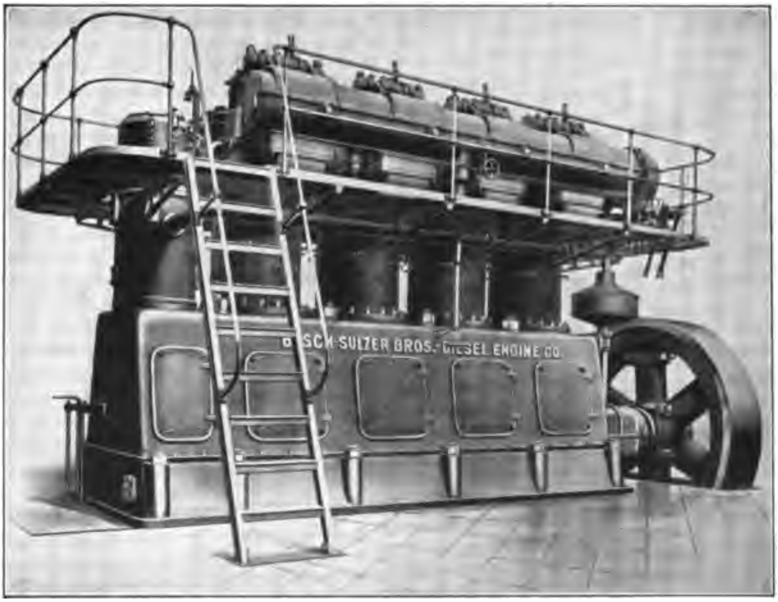


Fig. 110.—Four-cylinder Vertical Diesel Engine. Busch-Sulzer Bros.-Diesel Engine Company.

to the cylinder jacket by means of studs which pass through the entire height of the head. Fastened to the cylinder heads are also the supports which carry the valve lever shaft.

The **crank-shaft** is a one-piece mild steel forging. All cranks are in one plane, the two center cranks forming an angle of 180 degrees with the two end cranks.

The **bearings** consist of simple babbitt-lined shells, made in halves. They rest in cylindrical seats, integral with the bed-plate, and are kept in place by a rigid cap also bolted to the bed-plate.

The **connecting-rod** is of forged steel. The bearing at the crank end is made of cast-steel and is lined with babbitt metal. It is made in halves and is bolted to the T-shaped end of the rod. The upper end is closed and contains the bronze bearing for the wrist-pin.

The **piston** is made of cast-iron. In the upper part 6 or 7 grooves are provided for the piston rings, which are made of hard, wear resisting cast-iron. In the two larger sizes provisions are also made for water cooling the piston. The wrist-pin is made of steel, hardened and ground, and fastened in conical seats provided for this purpose in the piston.

At the flywheel end of the engine, and driven from the crank-shaft by a fully encased pair of screw gears, the vertical governor shaft is located. This shaft in turn drives the cam-shaft by means of another pair of screw gears. The cam-shaft is located alongside and on a level with the cylinder heads and fully enclosed in a cast-iron casing provided with hinged covers. In this manner it is possible to run the cams in oil.

The **valves** are operated by means of two-armed levers and rollers, the rollers being in close contact with the cams, and by means of the lever arms press on the spring-loaded valve stems, opening and closing the valves in accordance with the timing of the cams. The starting- and injection-valve levers are eccentrically supported on the valve lever shaft. By a simple turn of the handle both levers are brought in their correct position relative to the cams. For instance, for starting, the roller of the injection valve lever at no time comes in contact with its corresponding cam; while the starting-cam opens the valves at the desired instant and admits compressed air to the cylinder. This status is reversed for running on fuel.

The **fuel pump** is of the multiple type, *i.e.*, one plunger is provided for each cylinder, but they are all operated by means of one eccentric located on the vertical governor shaft. The pump is adjusted so that each working cylinder takes an even share of the load. Regulation is effected by the governor keeping the suction-valve open over a greater or lesser part of the working stroke of the fuel-pump plunger in accordance with a lighter or heavier load carried by the engine.

The **compressor** providing compressed air for starting and for injecting the fuel into the cylinder is of the vertical three-stage

type, and is driven directly from the crank-shaft by means of an additional crank at the end of the engine opposite the flywheel. Virtually the compressor has the appearance of a fifth cylinder on the engine. The air on its way through the compressor is cooled, and the moisture is abstracted in the receivers provided for this purpose.

A **forced lubrication system** is provided for in these engines. An oil tank is located below the engine room floor near the compressor end. It contains duplicate filters and a cooler in the lower part. After the system is filled, the oil can be used over and over again. The oil that has been collected in the lower part of the bed-plate runs into the top of the tank, through the filter, and then through copper cooling pipes. Here the suction pipe of the pumps is connected. The oil pump is located inside the bed-plate. It is of the rotary positive displacement type and is driven by gearing from the crank-shaft. The header from the pump is connected to each main-bearing. Suitably drilled holes in the crank-shaft and connecting-rod bring the oil to the crank-bearing and wrist-pin bearing, thus providing lubrication in these spots.

The rubbing surfaces in the working and compressor cylinders are lubricated from a multifeed pressure lubricator, forcing an adjustable quantity of oil at definite intervals into the cylinder at several points of the circumference at the same time.

A **water cooling-system** takes care of those parts of the engine which need cooling. While one branch of the system cools the cylinders, cylinder-heads and exhaust-valves, another branch is connected to the oil cooler, and from there the water flows to the compressor and thence to the exhaust-piping.

An entirely separate system is used for **piston cooling.** Water under pressure is injected through a nozzle into the upper part of the piston, from whence it overflows into a system of sliding tubes and is conducted to a funnel outside the engine frame.

In order to get at the valves and inspect the cam and roller gear, a stairway and platform is provided, which on the smaller units exists only in front of the engine, while on the two larger units it is carried all around the cylinders.

A feature of all sizes built is the compression relief gear, provided for the purpose of reducing the compression in the working cylinder when the engine is started up with compressed air.

Another feature of great importance, especially for the larger units, is the automatic injection air regulation. The air pressure with which the fuel is injected into the cylinder must be reduced with diminishing load on the engine to avoid knocks. This can be done by hand on the smaller engines where the load is not subject to great variations; but on larger units, with rapidly changing loads, as for instance street railway service, hand regulation would be inadequate; therefore, a device is provided on such units whereby the injection air pressure is automatically increased or decreased, the governor influencing a throttling valve located at the suction opening of the low-pressure compressor. Combined

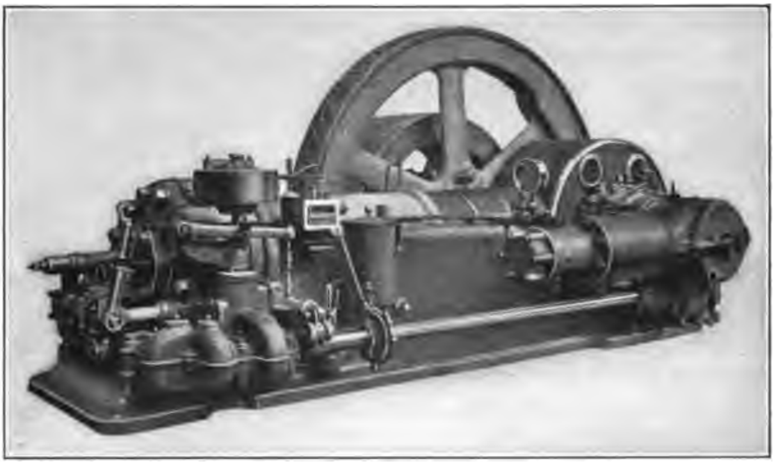


FIG. 111. — Single-cylinder Horizontal Diesel Engine. Snow Steam Pump Company.

with this regulation is a device to automatically adjust the lift of the fuel needle to the load. This is necessary to obtain close regulation at light loads.

The Snow Crude Oil-engine. — This engine, illustrated in Figs. 111 to 114, is built by the Snow Steam Pump Company of Buffalo, N. Y. It operates on the Diesel cycle and is made in both two- and four-cycle types, in sizes from 60 to 1500 horsepower.

It will be noticed that the main-frame and cylinder jacket are cast in one piece. The cylinder liner is then made in the form of a plain hollow cylinder and is forced into place, being held

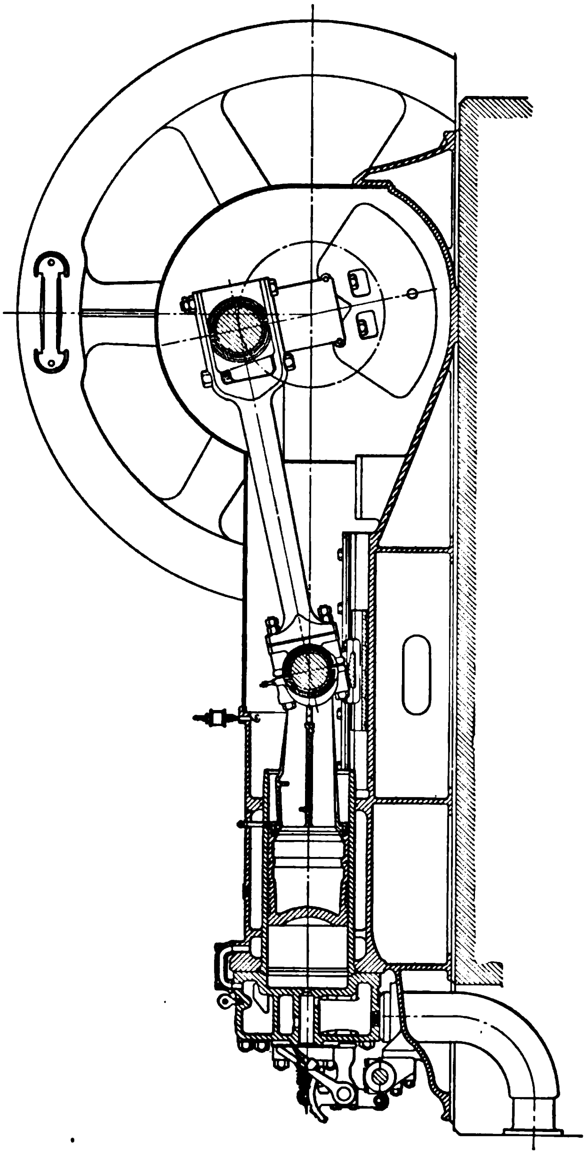


FIG. 112. — Longitudinal Section of Snow Four-cycle Diesel Engine.

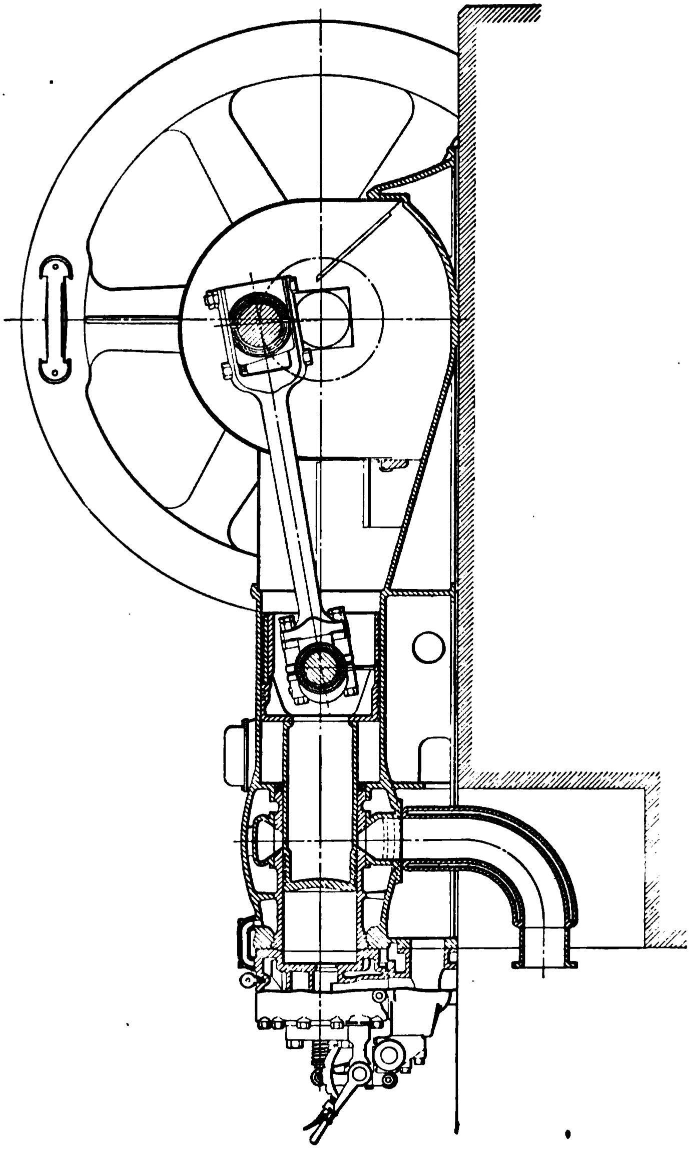


FIG. 113. — Longitudinal Section of Snow Two-cycle Diesel Engine.

in position by the cylinder head. In the **four-cycle** unit, shown in Fig. 112, the cylinder liner is of simpler construction than in the **two-cycle** unit, Fig. 113. The four-cycle unit has a crosshead, and the two-cycle unit has not. These two facts mark the chief mechanical difference between the two- and four-cycle units. In describing this engine, then, it will be necessary to describe only one type, and that will be the four-cycle.

The **piston** is of the usual trunk pattern, carrying seven rings. The piston proper is bolted to a long **crosshead,** which

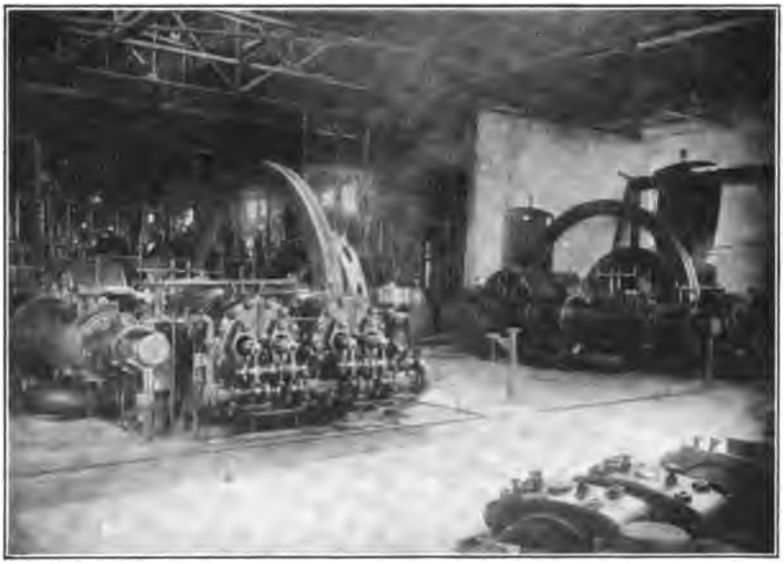


FIG. 113a. — Snow Single-acting Twin Diesel Engines Direct-connected to Generators.

has a guide of the marine or slipper type, formed in the frame. The shoe of the crosshead is babbitted.

The **cylinder head** contains the main inlet- and exhaust-valves, the injection-valve and the air-starting valve. These valves are placed in cages, any one of which may be removed without disturbing the others.

The **connecting-rod** is of the marine type at the wrist-pin end. The crank-pin end is formed by an open strap with a wedge adjustment to take up wear. The **shaft** is a solid steel forging with the counterweights bolted to the arms and secured by round keys.

The **cam-shaft** is driven from the main-shaft by spiral **gears.** At the end of the cylinder the short cross-shaft that carries the cams is geared to the side cam-shaft by bevel gears. The cam-shaft and cross-shaft are supported entirely by the frame so that the cylinder head may be removed without disturbing either one of them.

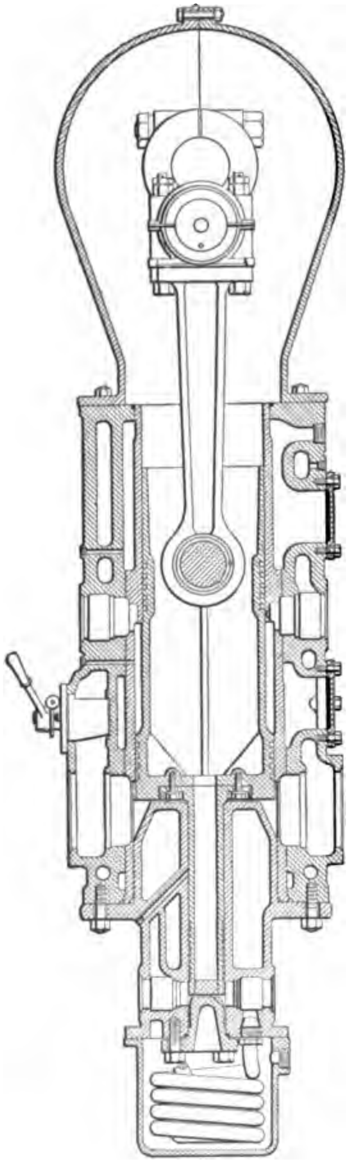


FIG. 114. — Longitudinal Section of Three-stage Air Compressor used on Snow Diesel Engines.

The **governor** is of the Jahn type and controls the stroke of the fuel pump through a wedge arrangement. This pump is simply a ground plunger operated on the discharge stroke by a lever and returned on the suction stroke by a spring. No packing is needed as the fuel is pumped into the spray nozzle when that part is not under pressure.

The **lubrication** of the cylinder is effected by a Richardson positive feed pump, which also furnishes oil for the valve-stems and compressors. Other bearings are oiled by a continuous oiling system including filter, pump and tanks.

The **air compressor** which furnishes high-pressure air for spraying the fuel into the cylinder is mounted on a pad on the side of the frame, and is driven by a drag crank on the end of the shaft. This compressor, which is illustrated in Fig. 114, furnishes not only the air for spraying but also that for recharging the starting tank. No high-pressure air-storage tanks

are required as the spraying air is pumped directly into the spray nozzle. This compressor is a three-stage unit for all engines of 250 horse-power and upwards. The first is the annular space around the small piston. The second stage is the annular space around the large piston, and the third, the space or cylinder in which the small high-pressure piston works. At the left end of the figure is shown the after-cooler, through which the air passes from the last stage. For engines less than 250 horse-power the compressor is two-stage.

These engines are started by compressed air. A lever operates in a quadrant marked for the starting, running and stopping positions. When the lever is shifted from the neutral to the starting position, low-pressure air is admitted to the cylinder through the air-starting valve, the spray valve is disengaged and the cam roller is shifted to one-half compression. After five or six revolutions the air in the pipe between the compressor and the spray valve is compressed to a sufficient pressure for spraying the charge of oil. The fuel injection pump is then thrown into engagement and the lever is shifted into the running position, which disengages the compression relief and the air-starting valve, and throws the spray valve into operation.

The two-cycle engine having the scavenging piston concentric with the motor piston is built in sizes up to 150 horse-power per cylinder. For engines larger than this a separate double-acting scavenging cylinder is provided. This cylinder is bolted to the main-frame on the side opposite the flywheel. The piston is driven by a crank on the end of the main-shaft.

In both types of the two-cycle engines the scavenging air is taken from a reservoir in the main-frame, and is admitted to the motor cylinder through two inlet-valves, the same valves that in the four-cycle are used for inlet and exhaust. The heads are, in fact, identical in the two types. The short time available for scavenging the cylinder makes it necessary to provide as much scavenging valve area as possible.

The Otto Crude Oil-engine.—This engine, Fig. 115, built by the Otto Gas Engine Works of Philadelphia, is of the four-cycle type operating on the Diesel principle. It conforms very closely to the engine built at the German works of the same company, the Gas Motoren Fabrik Deutz, Cologne-Deutz.

The **frame, bearings** and **water jacket** are all cast in one piece.

The **cylinder liner** is inserted in the water jacket in such a way as to allow for expansion. This construction, which is generally used in Diesel engines and all large gas-engines, is very good as it allows the liner to be made of a different grade of iron. It is a simple casting and is therefore more reliable and it may be easily replaced when worn.

The **piston** is of the regular trunk type furnished with many soft cast-iron packing rings. The **piston-pin** is of steel, hardened and ground, and held in the body of the piston by a key and set screw.

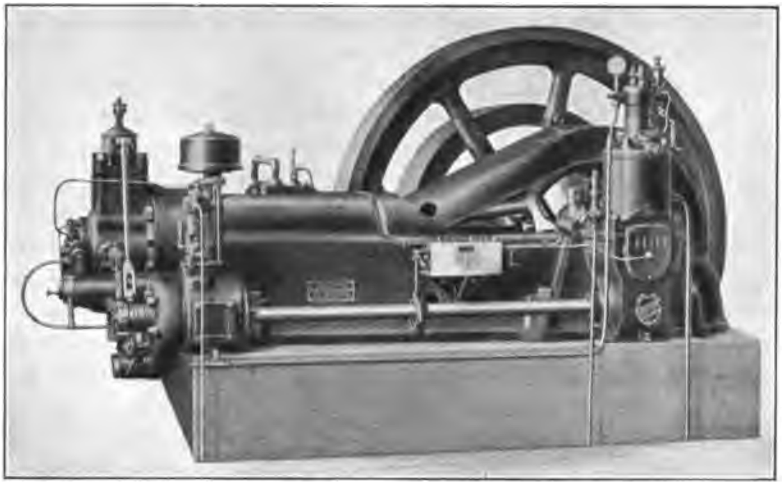


FIG. 115. — Otto Crude Oil-engine. Otto Gas Engine Works, Philadelphia, Pa.

The **connecting-rod** is of 0.20 carbon, open-hearth steel. The box at the piston-pin end is of phosphor bronze and at the other end of a special anti-friction metal. Both boxes are adjustable to take up wear. The **crank-shaft** is of open-hearth steel, forged from a billet and machined all over. The counterweights are fastened to the crank-arms. The bearings are of special anti-friction metal, oiled with ring oilers. The out-board bearing has a wedge adjustment.

The **valves** are placed in the cylinder head, the inlet being on top, the exhaust on the bottom and the fuel injection-valve on the side. The cam-shaft, supported entirely on the frame, is driven

from the main-shaft by spiral gears. This shaft has white metal bearings, oiled with ring oilers.

The **governor** is driven from the cam-shaft. The **fuel pump,** also driven from the cam-shaft, is actuated by cams. The governor controls the amount of fuel injected by this pump by varying the closing of the by-pass valve in the overflow chamber.

The **air compressor** which furnishes the injection air is shown at the right, directly over the crank-shaft. It is driven by a crank on the end of the shaft and is of two stages. The high-pressure air is delivered directly from the compressor to the fuel injection

FIG. 116. — Deutz Diesel Engine of 450 Horse-power.

valve, without an intermediate receiver. Air from the first stage is taken off and stored for use in starting the engine. The safety-valve on the first stage of the compressor is set at 220 pounds, and on the high-pressure stage at 1200 pounds.

A forced-feed **lubricator,** driven by an eccentric on the gear shaft, supplies oil for the cylinder, piston-pin, crank-pin and air compressor connecting-rod. Sight-feed oil cups are provided for smaller bearings.

The Deutz Diesel Engine. — This engine, illustrated in Fig. 116, is a four-cycle unit of 450 horse-power. It operates at

215 revolutions per minute. In general design it is not unlike some of the units already described. The cam-shaft is horizontal and about on a level with the tops of the cylinders. The three **valve arms** on each cylinder are plainly shown. They operate the inlet-, exhaust- and fuel admission-valves. The air compressor is horizontal — shown at the right-hand end of the engine, directly connected to the crank-shaft.

Fig. 117. — M. A. N. Diesel Engine of 2000 Horse-power.

The M. A. N. Diesel Engine. — The Maschinenfabrik Augsburg-Nürnberg A. G. manufactures a large line of Diesel engines at its Augsburg plant. One of these, illustrated in Fig. 117, has been placed in the plant of the Municipal Electric Works, Halle, Germany. It is a twin-tandem, double-acting unit of 2000 horsepower, one of the largest stationary Diesel engines ever constructed.

CHAPTER XVII

LARGE GAS-ENGINES

General. — It is rather hard to draw the line between medium and large-sized gas-engines. Medium-sized engines, however, are built in both vertical and horizontal units, while large engines are not built in vertical types. There is another difference that usually exists: large engines are made double-acting, while small and medium-sized units are almost invariably single-acting for stationary work. We might, if we have no objection to getting down to fine points, conceive of a single-acting engine made double-acting, and thus transferred from the medium into the large-sized class. It is certain that if we have two engines of the same bore and stroke, one single- and the other double-acting, the latter will be much larger than the former, not only in horse-power but also in weight and floor space required.

In taking up the subject of large engines for discussion, it will be impossible to go into great detail with even a few different makes of engines. It would be well to describe one or two in detail and take up distinctive features of others and this is the plan the author follows. This scheme will leave the reader with a good general idea of the large gas-engines made in this country, for there is a similarity that runs through all different makes, and this similarity is more marked in large engines than in smaller ones. The details that differ are usually the valve gear, governing system and details of construction. The latter, it will be impossible to point out in this chapter but they will be touched upon in the design.

The Cooper Engine.—This engine is built in single-tandem and twin-tandem, double-acting units from 200 horse-power up. All units work on the four-stroke cycle. The one shown in Fig. 118 is a single-tandem direct-connected to a natural gas compressor in the rear. The introduction of the high-pressure system of transporting natural gas has opened up a large field for gas-engines and many large units have been sold for this purpose.

The Frame. — The frame or bed-plate of this engine is cast in what is known as the Tangye rolling mill design. It is cast

with one bearing only, so that one side of the frame is high, of box section, while the other is lower. The illustration, Fig. 118, shows the low side. The main-frame extends back to the first cylinder, which is bolted to it. The **crosshead slide** is formed on the bottom of the frame, the top being open.

Cylinders.—The cylinders are made of close-grained, remelted charcoal iron. They are symmetrical one-piece castings, but the entire jacket is not cast with the cylinder. There is a ring left out of the jacket in the middle of the cylinder, leaving part of the jacket projecting from each end of the cylinder. This opening

FIG. 118. — Cooper Tandem Double-acting Gas-engine.

is closed by a **jacket band** in three segments. The bolts and flanges of this band may be seen in Fig. 118. The construction is shown in Fig. 192.

The cylinders are supported in the counter-bores of the **main bed-plate, distance-piece** between the cylinders and **tail-guide.** As the **pistons and rods** are carried by the **crossheads,** the cylinders support only their own weight.

Pistons. — The **pistons** are cast in one piece without any internal ribs. Each piston is fitted with three **rings** made in segments, having malleable iron **keepers** at the joints. These packing rings are held out against the cylinder walls by steel

springs. The pistons are forced on the rod against a shoulder and held there by a flush nut. Provision is made to insure circulation of water through the piston.

Piston-rods.—The **piston-rods** are nickel steel forgings drilled from both ends for water circulation. They are fastened to the main, intermediate and tail-guide crossheads by nuts. All rods are of the same size and interchangeable. The rods are fitted with **metallic packing** of the sectional type, consisting of segmental cast-iron rings placed along the rod in series. The packing includes two **fire rings** adjacent to each combustion chamber, which are compressed by external snap rings, and the remaining rings in the series are held in contact with the rod by garter springs.

Crossheads. — The **main** and **intermediate crossheads** are of open-hearth steel. The main crosshead is forked, and is connected to a flat babbitted shoe by means of a cylindrical joint, in which vertical and rolling adjustments may be made. The main **crosshead-pin** is of high carbon steel, clamped into both crosshead cheeks.

The **intermediate crosshead,** with its cylindrical shoe, acts as a coupling for the two piston-rods and admits cooling water, which is made to enter at the ends of the rod. The **tail-guide crosshead** also has a cylindrical shoe provided with means for centering and adjustment.

Distance Piece and Tail-guide. — The distance piece between the cylinders rests on machined ways on a **base-plate** to allow for expansion and contraction of cylinders. This casting also provides two bored guides, one for the intermediate crosshead and one for the water crosshead, to admit cooling water to the rods and pistons through tubes that telescope with water chambers. The base-plate underneath supports bearings for a lay-shaft. The **tail-guide** is a cylindrical casting and provides a slide for the rear crosshead. This casting is cored out to allow the escape of cooling water from the rear piston-rod.

Connecting-rod. — The **connecting-rod** is made of open-hearth steel and is of solid end construction as shown in Fig. 171. Both ends are provided with babbitt-lined shells and wedge adjustment. The adjustment at the crank-pin end tends to shorten the length of the rod while the adjustment at the other end tends to increase the length.

Crank and Main-shaft. — The side-crank construction is used in this engine, as in practically all large American gas-engines and steam-engines. The **crank** is a high-carbon open-hearth steel casting having the **counter-weight** and **crank-pin** cast solid with it. The crank is forced onto the main-shaft under hydraulic pressure, and is keyed into place.

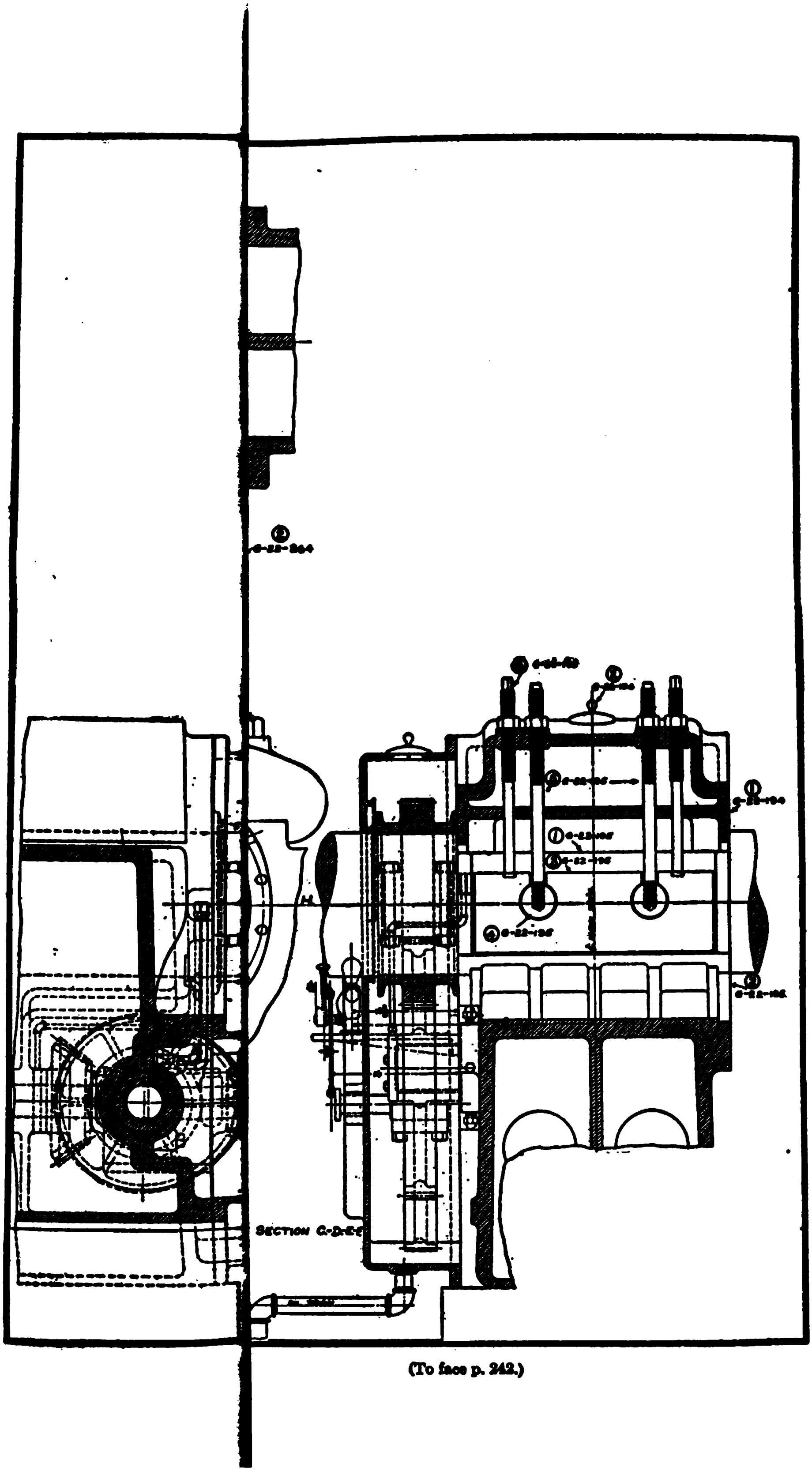
Valve-gear. — The **cam-shaft** is driven from the crank-shaft through three spur gears and one pair of bevel gears. These gears are enclosed in a case and run in oil as shown by Fig. 119. The cam-shaft proper is supported on bearing brackets cast on the engine base. It will be noticed that the shaft is held between two collars at the bevel gear. It is free to move longitudinally at all the other bearings, however, — a necessary precaution to allow for expansion and contraction. This precaution is even more necessary when the bearing supports are on the cylinders.

The **valves** are operated by **eccentrics** and rolling levers as shown in detail in Fig. 120. The **exhaust-valve** is water cooled. Both valves have a constant lift. **Governing** is accomplished by rotating the sleeve which encloses the inlet-valve spring. This sleeve has ports which register or cover as the load increases or decreases. These sleeves are under the control of a Jahn centrifugal governor.

This system gives constant-quality regulation with variable quantity. There is a means, however, of adjusting the sleeves by hand in order to get the proper mixture for any kind of gas.

Ignition. — The **make-and-break** system is used and two igniters are used in each combustion chamber as illustrated in Fig. 121. The **cam** which operates the igniters is on the **igniter-shaft** which is above and parallel to the cam-shaft as shown in Fig. 118. As the finger drops off the high point of the cam, it is pulled back into place by the **coil spring** at the finger pivot. This motion is transmitted to the two push rods and by them to the moving electrode. The moving electrode is made of cast-iron, the stationary one of soft steel. Both are insulated from the body of the igniter by lava sleeves and mica washers.

Ignition may be varied on all igniters, at once, by the movement of one central lever, or each igniter may be adjusted individually.



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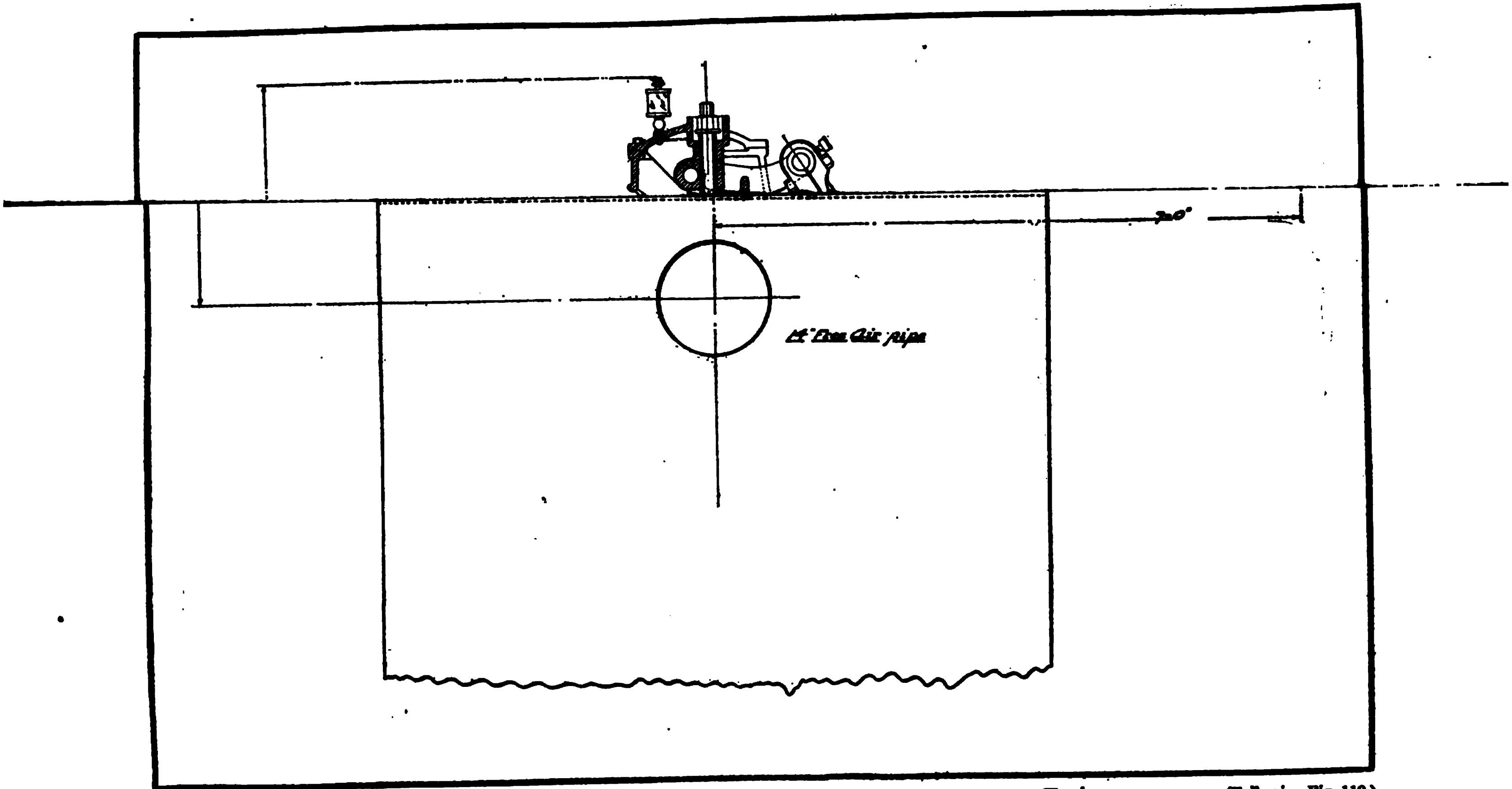


FIG. 120. — Section Through Combustion Chamber of Cooper Gas Engine. (Following Fig. 119.)

Lubrication. — The engine is lubricated by a **pressure** oiling system in which uniform pressure is maintained at all of the engine oil feeds by a pump under the control of a **weighted**

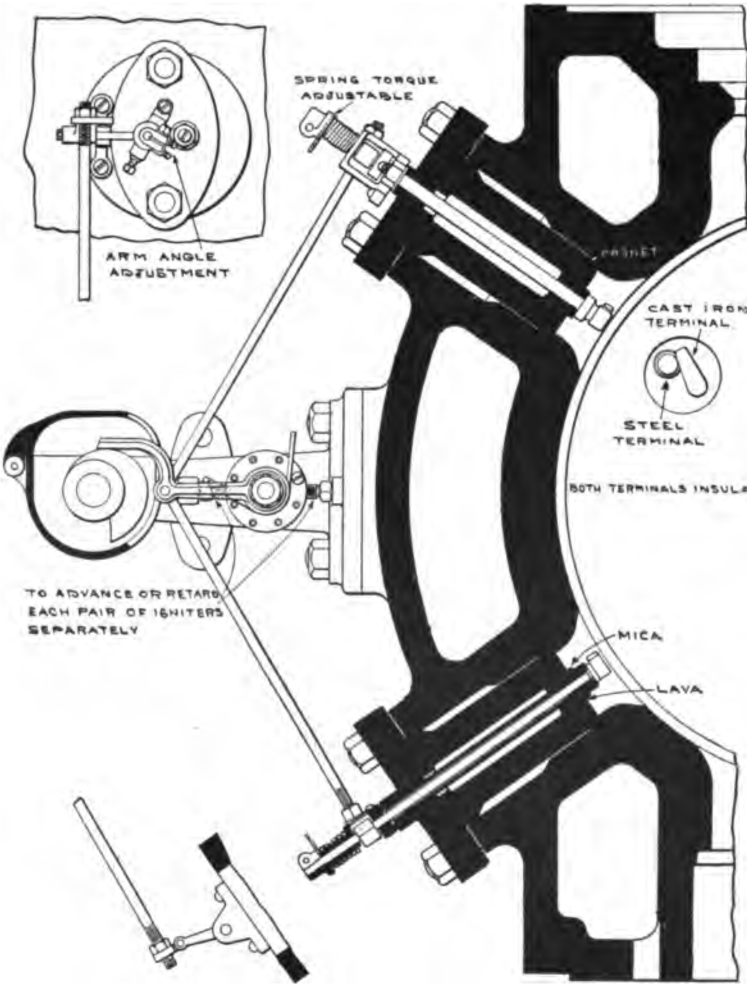


FIG. 121. — Ignition System. Cooper Gas-engine.

valve, which automatically returns any excess oil to the filter through a by-pass. The rotary oil pump, which supplies oil from the filter to the pressure feeds, is driven from the main engine. The oil, after being used, is caught and returned by

gravity to a settling tank at the side of the engine foundation. From this tank the oil flows into a filter, where it is cleaned to make it suitable for using again. The cylinders and piston-rods are lubricated by multiple force feed pumps, driven from the igniter shaft. Oil is admitted to the piston-rods through the metallic packing.

Starting.—The air, gas and water valves are placed at a central point, all within reach of the operator. To start the engine the compressed-air and gas valves are opened. The compressed air "cranks" the engine and after a few revolutions the air is automatically replaced by the explosive mixture when ignition

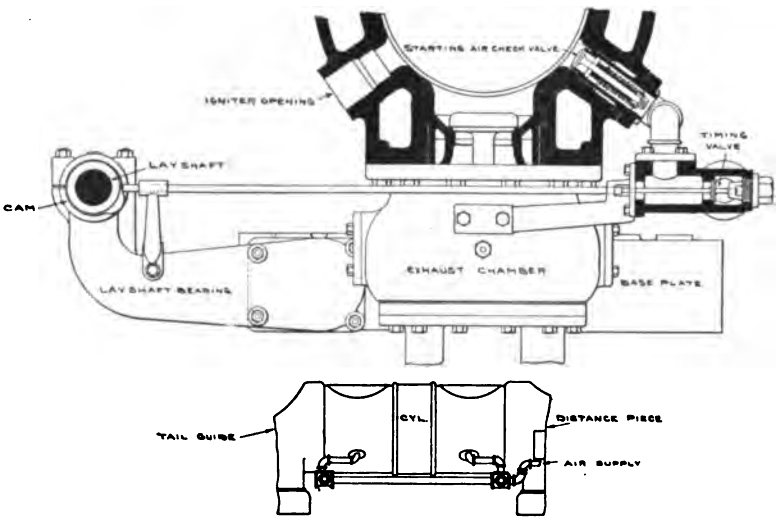


FIG. 122. — Air Starting Gear. Cooper Engine.

begins in the cylinders. This result is accomplished by four cams on the cam- or lay-shaft, each cam operating a poppet valve leading to the combustion chamber of the cylinder. This is shown in detail in Fig. 122. When the compressed-air throttle is opened, air is admitted on top of the **timing valve** and holds it against the seat. The cam forces this valve open at the proper instant, and air is admitted to the cylinder through the check valve. This occurs on the explosion stroke but before explosions begin. When explosions do take place, the pressure in the cylinder is higher than the pressure of the compressed air, the check valve is held on its seat and no air enters. The

operator then closes the compressed-air valve and the timing valve drops away from its cam.

The Buckeye Engine. — The double-acting tandem four-cycle type of Buckeye engine is illustrated in Fig. 123. This type is made in sizes from 250 to 3000 horse-power or in the twin type from 500 to 6000 horse-power. The same company, the Buckeye Engine Co., Salem, Ohio, furnishes single-cylinder double-acting engines, and single, two- and four-cylinder single-acting engines. All are made horizontal. The engine illustrated has cylinders cast with the non-continuous water jacket described before. The different parts of the engine including main-frame, cylinders, distance-piece, tail-guides and supporting bed-plates are clearly shown.

The **pistons** are made in halves, split at right angles to the axis of the rod and clamped together. Six packing rings are used. The **piston-rod** is made in two parts and joined at the intermediate crosshead. Each part is drilled through from end to end. The **main crosshead** is a steel casting, threaded to receive the piston-rod. It is also split and clamped

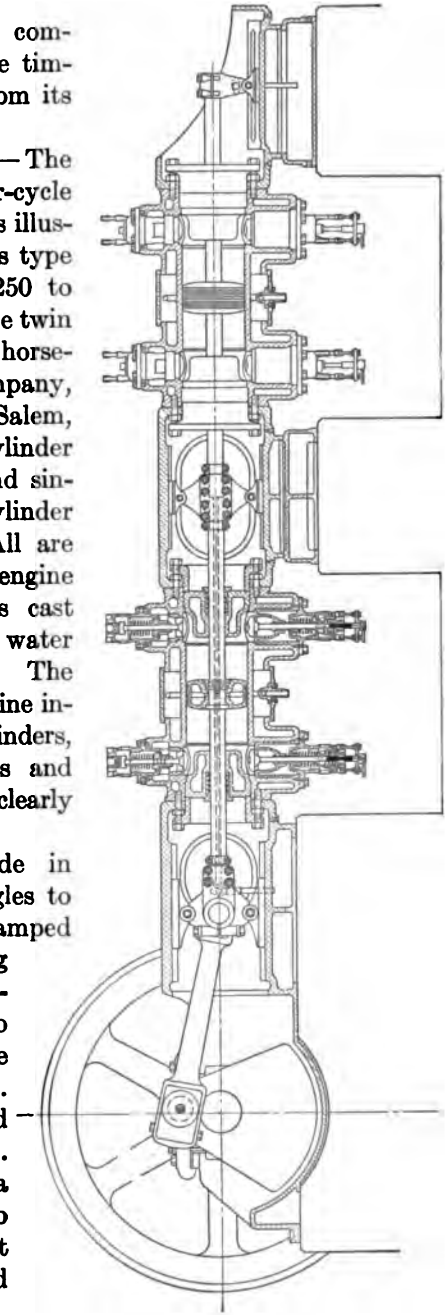


Fig. 123. — Longitudinal Section of Buckeye Four-cycle Engine.

on the rod by means of through bolts. The crosshead **pin** is straight and is secured in the crosshead by means of clamping bolts. The **shoes** are steel castings and have a swivel connection to the body of the crosshead, and are adjusted for wear by eccentric bolts, which are clamped after adjustment. The **intermediate crosshead** has the same swivel shoe connection as the main crosshead, and also serves as a clamping nut to connect the two sections of the rod. The **tail-rod crosshead** consists of a simple steel casting provided with one swivel shoe and has the same adjustments as the other crossheads.

The **connecting-rods** are forged from open-hearth steel. In double-acting engines the rod is solid at both ends and has bronze boxes at the crosshead end with wedge adjustment. At the crank-pin end the boxes are babbitted and also provided with wedge adjustment. In **single-acting** engines the crosshead end is made solid, but the crank-pin end in this case is of the marine type with babbitted boxes.

Each pair of **valves** on this engine is driven by a single eccentric. The eccentrics are clamped to the **lay-shaft.** They are turned spherical on the periphery and are adjusted by means of a clamped joint. The exhaust valve is water cooled. The lay-shaft is placed in a horizontal plane passing through the center of the engine. It is driven by a set of steel gears, one of which is mounted on a drag shaft that is rotated by the main crank-pin through a universal joint. This drag shaft, which is shown in detail in Fig. 124, also carries a bevel gear which drives the governor. These two pairs of bevel gears are entirely enclosed and are oiled automatically. The drag crank serves for carrying lubricating oil to the main crank.

The regulating system of this engine was described in Chapter XI.

The **ignition system** is designed to operate with either jump or make-and-break spark. The double-acting engines are provided with two or more igniters in each combustion chamber and can be operated with high- and low-tension ignition simultaneously. The engine is usually equipped with an electromagnetic make-and-break system, made by the makers of the engine. Current from a storage battery, magneto or generator operates the igniter mechanism by means of an iron-clad electromagnetic hammer. The circuit is periodically completed and

the time of ignition controlled by a " timer." This timer is provided with three adjustments: First, automatic control directly by the governor which advances the ignition on light loads when the charge is lean; second, hand adjustment of all igniters simultaneously; third, hand adjustment of each igniter independently while the engine is running.

The **cooling water** enters the piston-rod at the intermediate crosshead and leaves the rod at opposite ends, draining through slots in the bed and tail-rod guide. Each cylinder, cylinder head, exhaust-bonnet and exhaust-valve is cooled and has its own independent water circulation.

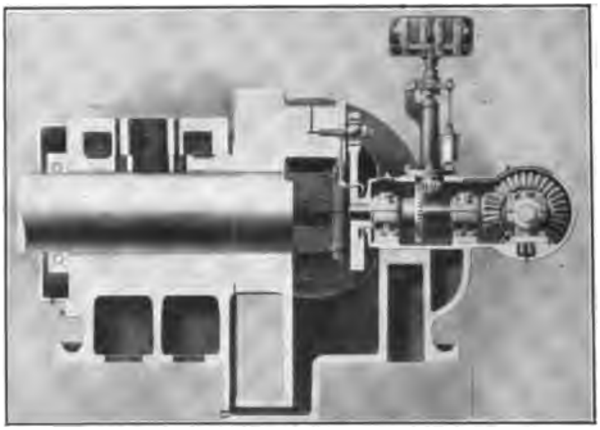


Fig. 124. — Cam-shaft Gears. Buckeye Gas-engine.

The Mesta Engine. — This engine, built by the Mesta Machine Company, Pittsburgh, Pa., is shown in Figs. 125 and 126 and in detail in Fig. 127. Among the distinguishing features of this engine are the valve-gear and the method of attaching the rods to the crossheads. The inlet-valve and governing mechanism are enclosed in a casing which extends the length of the cylinder. Gas and air are admitted at the center of this casing and distributed through passages in both directions of the two inlet-valves. In these passages are butterfly valves which are under the control of the governor and which determine the amount of gas and air drawn into the cylinder. The connections are shown clearly in Fig. 127.

The peculiarity of the **crosshead** and **rods** is that the rods are fixed in the crossheads by means of taper cotters or keys.

FIG. 125. — Mesta Gas-engine Direct-connected to Alternator.

The **igniters,** operating mechanism and wiring are well illustrated in Fig. 127. Fig. 126 illustrates one of the latest Mesta gas-

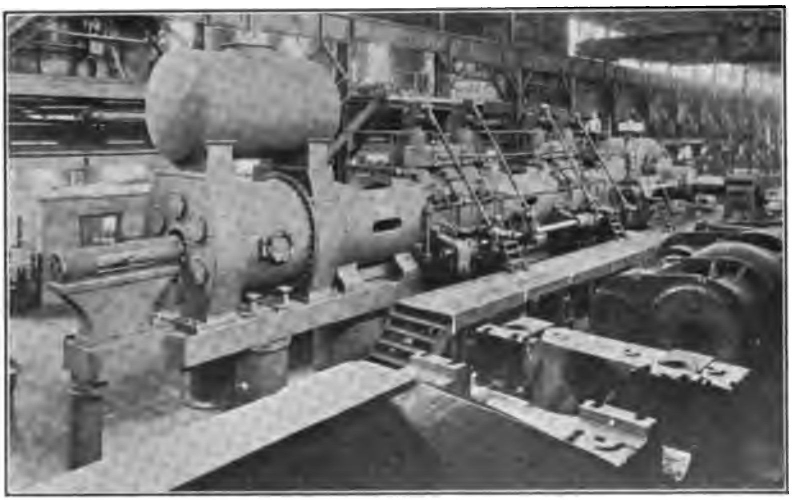


FIG. 126.

driven blowing engines for blast-furnace work. The fuel is blast-furnace gas.

The Snow Engine. — This engine is built by the Snow Steam Pump Works, Buffalo, N. Y., in sizes above 500 horse-power. It is of the four-cycle type and is built in either single- or twin-tandem double-acting units.

The **cylinders** are made of soft high-silicon iron, cast in halves. Each cylinder is fitted with a hard, close-grained, low-silicon cast-iron **liner** forced into the cylinder against a shoulder.

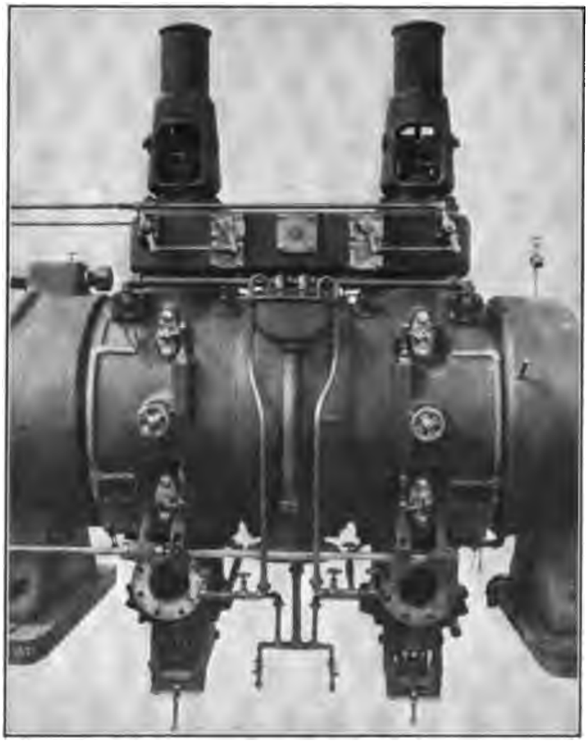


Fig. 127. — Cylinder Detail. Mesta Gas-engine.

A **jacket band,** acting as an expansion joint, is provided with water passages for carrying away the jacket water from the top of the cylinder. The cylinders are supported on four feet, which rest on tongued and grooved surfaces on a bed plate extending throughout the whole length of the engine. It will be noticed that the combustion chamber is on the side of the cylinder, a peculiarity of this engine.

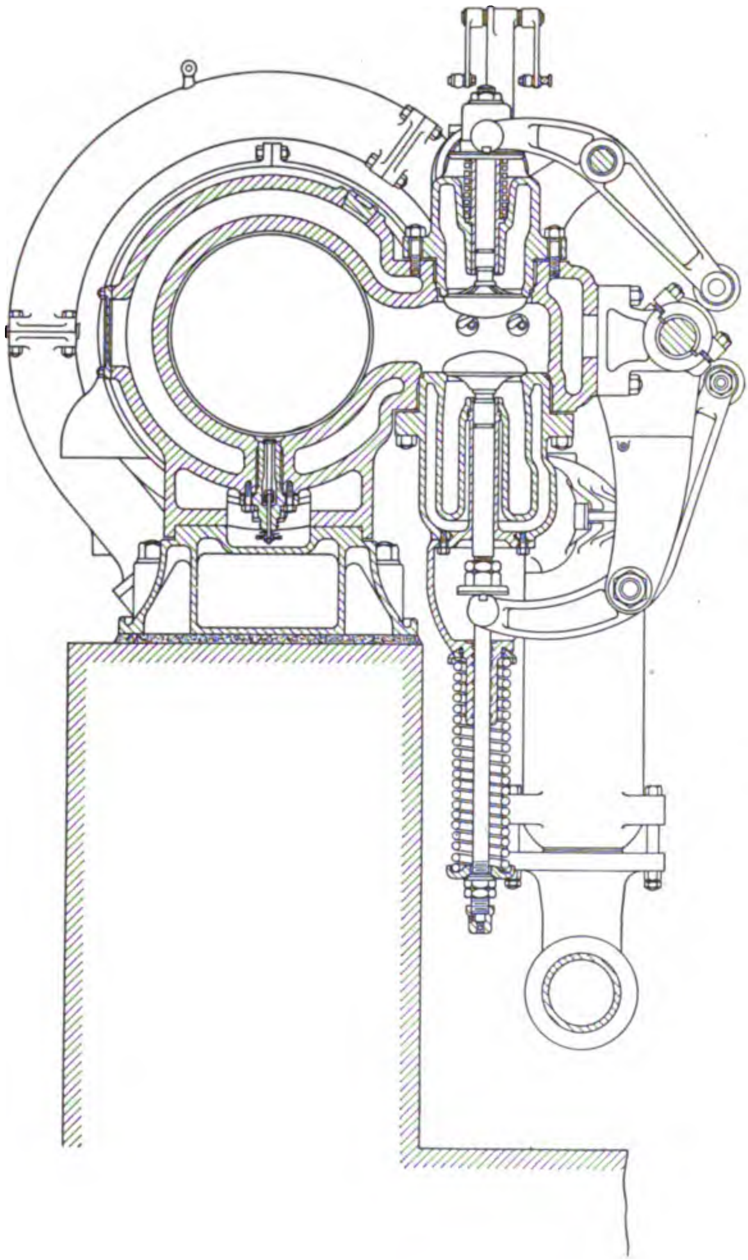


FIG. 128. — Section Through Combustion Chamber. Snow Gas-engine.

The **piston-rods** are of high carbon steel. The **pistons** are of cast-iron and carry six cast-iron snap rings of the double lap-jointed type. The piston-rods, cylinders and heads are all interchangeable.

The **main-frame** is fitted with crosshead top and bottom guides, the upper guide being formed in a loose, removable block; the guides are bored from the center, however, and the loose top piece provides for an easy removal of the inboard piston. Quarter-box **bearings** are used, the side shells being adjusted by means of wedges, and the lower shell being easily removed after the weight of the shaft is relieved.

The **connecting-rod** is of solid end construction; the boxes are of open-hearth steel, faced with babbitt metal and adjustable by means of wedges. The **crank-shaft** is of side-crank construction. The crank is of open-hearth steel, made separate from the shaft which is of hydraulic forged open-hearth steel.

The **valves,** illustrated in Fig. 128, are both operated from the same cam by simple levers. The exhaust-valve is water cooled. The **governor** is connected directly by means of reach rods to the regulating mechanism which consists of simple balanced poppet valves serving as both mixing and throttling valves, and which are so arranged that each cylinder can be adjusted separately. The governor is provided with a device which reduces the cylinder charges in starting until the engine is up to speed, thus avoiding heavy shocks due to full charges at low speed.

There are two **igniter plugs** in each combustion chamber. These are both operated from the same cam as shown in Fig. 129. As the tappet drops off the cam it is forced quickly to the right by the large spring. The tappets strike the moving electrodes and separate the points in the cylinder, causing the spark. The end of the tappet rod near the cam may be raised or lowered by means of the vertical screw and lock-nut, thus retarding or advancing the spark.

A **starting column** is provided with this engine on which will be found the hand wheels controlling the jacket water, air for starting, gas throttle, the oil pressure from the overhead tanks for the bearings and the switch controlling the igniters. A compressed-air distributing valve at the end of the cam-shaft distributes air for starting to the proper combustion chambers

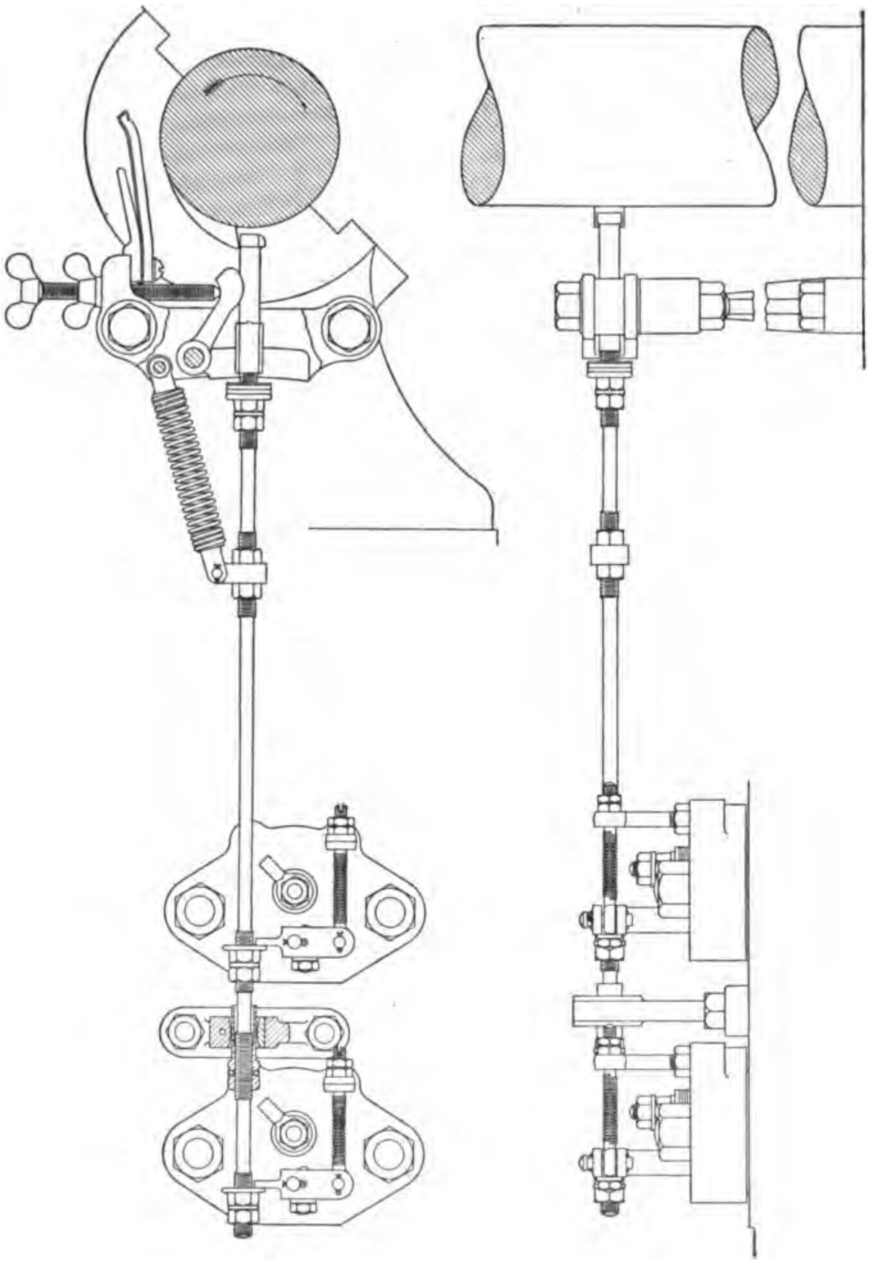


FIG. 129. — Make-and-break Ignition Device. Snow Gas-engine.

in turn. When explosions begin to take place this air-valve is thrown out of gear.

The Nuremberg Engine. — This engine, manufactured by the Maschinenfabrik Augsburg-Nürnberg A. G., is illustrated in Figs. 131 to 133. It is of the four-cycle type, double-acting, with cylinders arranged either single- or twin-tandem.

The **bed-plate,** or main-frame, is a box casting resting on the foundation for its entire length. It contains two **main-bearings,** the **crosshead slide** and the flange for making the connection

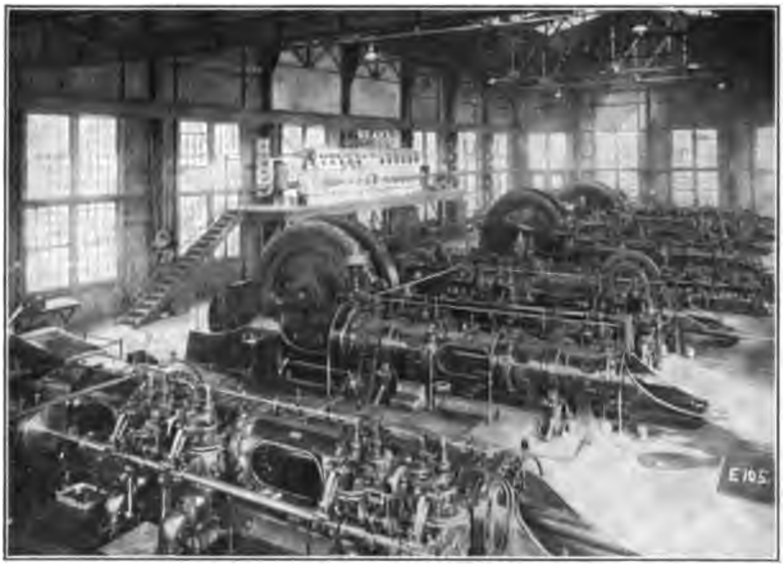


FIG. 130. — Four 800-horse-power Twin-tandem Snow Gas-engines using Producer Gas for Fuel. Engines are Driving 60-cycle Alternators.

to the first cylinder. The crosshead slide and cylinder flange are bored at the same time to secure good alignment. The **main-bearings** have cast-steel shells of the four-part type lined with babbitt metal. It will be noticed that the main-frame carries two bearings instead of one, the latter being the common American practice. This is because the German engines are built with a **center** crank shaft instead of the **side** crank shaft which is prevalent in America.

The **cylinders** of the Nuremberg engine are made of cast-iron, the jacket and liner being cast integral with the jacket

continuous and not divided in the middle as are the jackets of the American engines. The **valve pockets** in this engine are formed in the cylinder proper and not in the head. The inlet-valve is on the top and the exhaust-valve on the bottom. The cylinders are connected by the usual distance piece which carries a slide at the bottom for the intermediate crosshead. At the rear of the back cylinder there is also the usual slide for the tail crosshead.

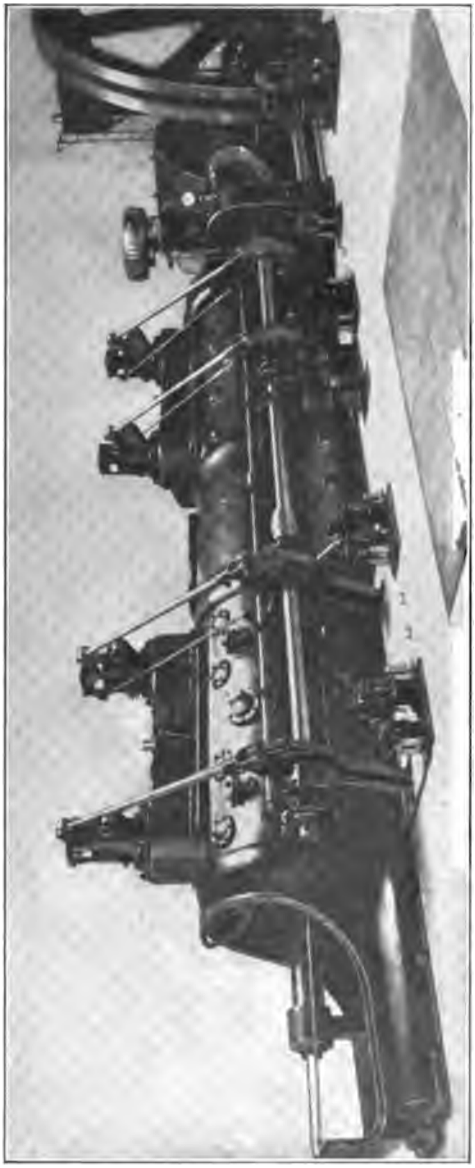


FIG. 131. — 2500-horse-power Nuremberg Gas-engine.

The **connecting-rod** is of the marine type at both ends. The boxes are of steel, lined with babbitt metal. The **piston-rods** are of crucible steel, hollow and water cooled, and are machined with a slight upward camber to allow for the weight of the rod and piston. The **pistons** are of cast-steel and are water cooled. The **rings** are of cast-iron, sprung into place. The **crank-shaft** is of mild steel.

Valve-gear. — At each cylinder end there is one combined

mixing- and inlet-valve *A* and *B*, Fig. 133, on the top and one exhaust-valve *C* on the bottom of the cylinder. The mixing-valve *A* regulates the gas and air by means of the air slide *D* rigidly connected to the gas-valve. The inlet-valve *B* is of the ordinary mushroom type and regulates the quantity of mixture admitted to the cylinder. The three valves *A*, *B* and *C* are operated by a system of rolling levers *E* and *F*, which derive their motion from an eccentric *G*, at each end of each cylinder.

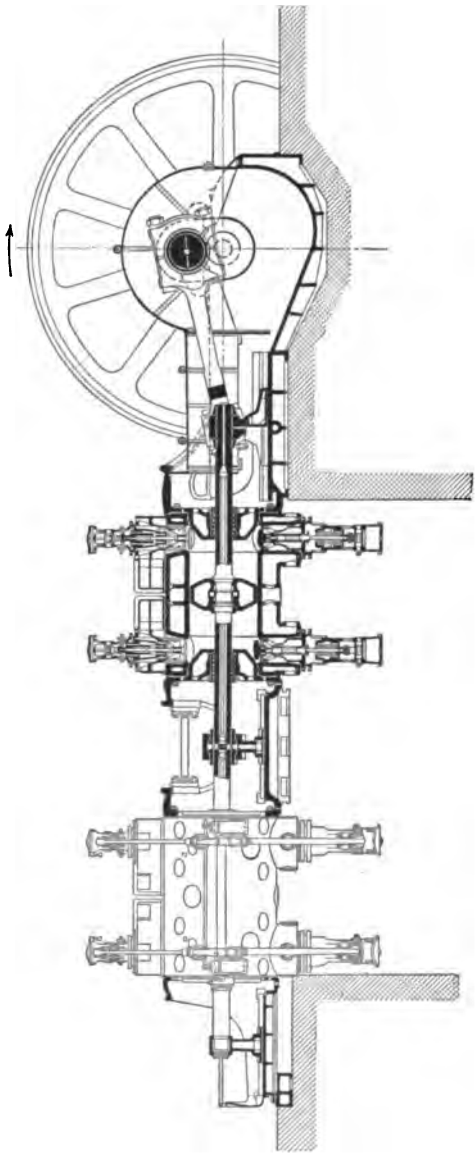


FIG. 132. — Longitudinal Section. Nuremberg Gas-engine.

The **governor** alters the lift of the combined mixing- and inlet-valve by moving a die *H* which is inserted between the upper and lower inlet valve rolling lever *E*, and in this manner the quantity of mixture drawn in is suitably proportioned to the required output. The composition of the mixture is practically the same for all loads, after the valves have once been set for the most efficient combustion according to the kind of fuel used. If the quality of gas changes,

then the ratio of air to gas can be readily changed by hand, while the engine is running, by turning the air slide valve *D* by means of a lever. The lay-shaft, that to which the valve eccentrics are keyed, is driven from the main-shaft by means of screw gears.

The **ignition** apparatus consists of a small accumulator battery, a contact apparatus or automatic switch, contact breakers and ignition plugs. The latter are provided in duplicate in each combustion chamber. The switch gear is mounted on the lay-

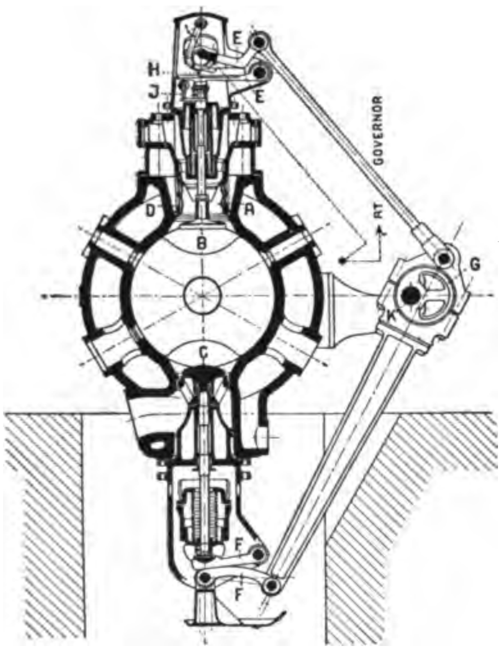


Fig. 133.—Section Through Combustion Chamber. Nuremberg Gas-engine.

shaft and so distributes the current from the storage battery that at the moment of ignition the current flows through its proper contact breaker and plug.

The **contact breaker** consists of a small electric motor, which, being under current, causes an armature to move, and the armature in turn pushes against the movable electrode in the spark plug. The point of ignition can be adjusted simultaneously for all eight plugs of a tandem engine by a hand wheel on the switch gear while the engine is running.

Lubrication. — Oil is forced into the cylinders, stuffing-boxes and exhaust-valve spindles by means of an automatic force-feed multiple-plunger pump, each feed having its own separate plunger. These pumps are driven from the valve-gear eccentrics. The lubrication of the main-bearings, the crosshead slide and the connecting-rod bearings is effected by the flush system from a tank placed some distance above the engine. As the oil drips from the bearings it is caught in the grooves of the bed-plate, whence it flows to a tank in the basement. Here it is filtered, after which it is pumped back into the elevated tank to be used over again.

FIG. 134. — Koerting Four-cylinder Single-acting Gas-engines Direct-connected to Alternators.

The bearings of the lay-shaft are of the ring-oiled type, while the pins and eccentrics of the valve-gear are lubricated by compression grease cups.

Cooling. — The cylinders, cylinder-heads, exhaust-valves, pistons and rods are cooled separately by individual water lines. The return pipes are all arranged close together on one side of the engine, and pour the waste water into a common open funnel. Each outlet pipe is fitted with a thermometer and valve, so that the temperature on each part may be regulated independently. One main stop-valve allows the water to flow to all parts of the

engine so it is impossible for the operator to neglect any one part. The pressure of the water for the cylinders and exhaust-valves is about 15 pounds gauge, but for the rods and pistons a pressure of 80 pounds is used. For the latter pressure a pump is provided which is driven direct from the main-shaft.

Koerting Engine. — In Fig. 134 is illustrated a type of engine prominent in Germany but almost unknown in America. It is a double-twin, single-acting, four-cycle Koerting engine, direct-connected to fly-wheel type alternators. The general construction is much the same as the double-acting engines already described, except that trunk pistons are used and two cylinders are side by side instead of end to end. The two cylinders together form what is known as the "drei-lager" or three-bearing engine. The crank-shaft for each pair of cylinders has three bearings, one between the cranks and one outside each crank, hence the name.

CHAPTER XVIII

THE HUMPHREY GAS PUMP

The **Humphrey** gas pump was invented by Herbert A. Humphrey of England. Its outward appearance has no resemblance to an internal combustion engine; it is only in operation that the two are similar.

For heads up to 50 feet the gas pump is very simple — a piece of pipe with a stand-pipe at one end and a combustion chamber at the other, the latter being surmounted by a few simple fittings. Fig. 135 is a diagram of the simple pump. It consists of a conical

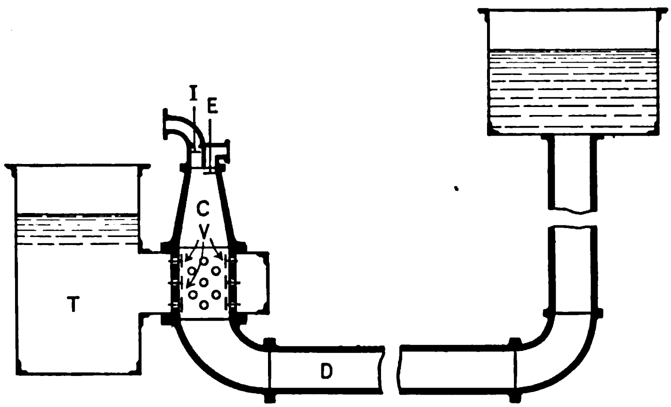


FIG. 135. — Diagram of Simple Humphrey Pump.

explosion chamber *C*, fitted at the top with inlet-valve *I*, the exhaust-valve *E*, and a scavenging-valve not shown. A simple interlocking gear is arranged between these valves by means of which, when valve *I* has opened and closed, it is locked in the closed position and valve *E* is released. When valve *E* has opened and closed it is locked and valve *I* is released.

Imagine a charge of gas and air compressed in the top of chamber *C* and fired by a spark plug which projects through the top of the casing. All the valves are shut when explosion occurs, and the increase in pressure drives the water downward in the

pump and sets the whole column of water in the discharge pipe in motion. The column of water attains kinetic energy while work is being done on it by the expanding gases, so that when these gases reach atmospheric pressure the column of water may be moving at, say, 6 feet per second. The motion of this column of water continues until the pressure behind it falls far enough below the atmosphere to open the exhaust-valve *E* and the water-valves *V*. Water rushes in through these water-valves and follows the moving column in pipe *D*, and at the same time it rises in chamber *C* in effort to reach the level in the suction tank.

When the kinetic energy of the moving column has been expended in forcing water into the high level tank, it comes to rest and, there being nothing to prevent a return flow, the column starts to move backward toward the combustion chamber and continues with increasing velocity until the water reaches the level of the exhaust-valve, which it shuts by impact. A certain part of the burned gas is trapped in the cushion space beneath the inlet-valve, and the energy of the moving body of water is expended in compressing this burned gas to a greater pressure than that due to the static head of the water in the tank *T*. A second outward movement of the column now begins and when the water reaches the level of the valve *E*, the pressure in the compression space is again atmospheric and further movement of the water opens the valve *I*, which has been released by the interlocking device, thus drawing in a fresh charge of gas and air. Once more the column of water returns under the pressure of the elevated tank and compresses the charge of gas and air, which is then ignited to start a new cycle of operations.

Ignition. — Ignition is timed by an apparatus resembling an ordinary engine indicator, which closes the electric ignition circuit at the point of maximum compression, and an ordinary small battery, trembler coil and spark plug, such as are used in automobile work, are employed.

Starting. — In starting the pump for the first time compressed air is allowed to flow into the combustion chamber until the volume of the air thus introduced is slightly larger than the usual charge. The exhaust-valve is then suddenly opened by means of a hand lever, and the escape of the compressed air permits a movement of the water column which gives the cushion and suction strokes, and so draws in a fresh combustible charge, which, on ignition,

starts the pump working regularly. If the pump is stopped when working regularly, it always stops with a fresh charge in the combustion chamber, under pressure, so that the only operation necessary for starting is to close the switch on the ignition circuit. The pump can therefore be started and stopped from a switchboard.

Two-cycle Pump. — The Humphrey pump described in the preceding articles works on the four-cycle principle, requiring four strokes for completion of one cycle. The time required for the completion of a cycle is much greater than in an ordinary engine and is not governed in design except by the length of the "play pipe" *D* in Fig. 135. Fig. 136 shows an indicator card taken from one pump in which the time per cycle was about 5 seconds, or actually the rate was 12.5 cycles per minute.

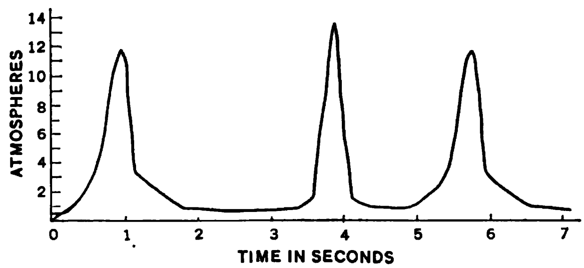


FIG. 136. — Indicator Diagram from Humphrey Pump.

In Fig. 137 is illustrated a double-cylinder pump which draws water twice each cycle, and, by means of the air chamber called the "intensifier," placed between the combustion chamber and a set of delivery valves, will deliver water against a head which is independent of the mean pressure of explosion. *A* and *B* are the two combustion chambers, *C* and *D* two water valve boxes, *E* the connecting chamber and *F* the discharge pipe. In the top of *A* and *B* are fitted the usual inlet- and exhaust-valves, but in this case the exhaust-valve is set near the top of the chamber as practically no cushion is wanted. The pump works with a drowned suction, being placed inside the tank *T*, the water supply being through *M*. Starting with a compressed combustible charge in *A* and with *B* full of water, explosion occurs and drives the water in *A* downward and outward, setting the whole column of water in *F* in motion. When the gases in *A* have expanded nearly to atmospheric

pressure, the water column in B is no longer supported and, falling by its own weight, draws in a fresh combustible charge which is throttled to prevent the charge being too large. The inlet water coming in through the valves follows the outward moving column and also rises in chamber A, to expel the burnt-products up to the level of the water in tank T. The column in the discharge pipe, having come to rest, now begins to return, and, rising in chamber A, expels the burnt-products and closes the exhaust-valve. The energy of the moving column is now expended in compressing the new combustible charge in B, whereupon ignition in B starts a new cycle with functions of A and B reversed.

Coming to the intensifier, we have the small air chamber G and the larger air vessel K with a valve box J. The small air vessel G is fitted with an adjustable pipe H, open to the atmosphere and

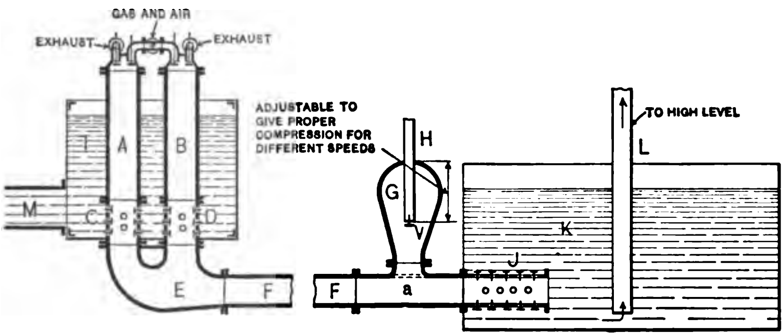


FIG. 137. — Humphrey Two-cycle Pump.

closed at the bottom by a valve which opens under its own weight and closes under the impact of the water on it. When the explosion occurred in chamber A the water level in the small air vessel was, say, a, and while rising from a to V the energy of the expansion in A is wholly used in giving kinetic energy to the water column. As the water strikes V it shuts this valve and then compresses air in G until the pressure equals that in the high-pressure air vessel K. The valves in J now open and the energy in the water column is expended in driving water through these valves into K. When the water column stops moving, the valves in J close and the energy of the compressed air in G drives back the water column, exhausts the burnt-products from A, and compresses the fresh combustible in B. Thus everything is ready for the next cycle.

It will be seen that by raising the pipe *H* the quantity of air compressed in *G* is lessened, and by lowering *H* the quantity is increased. Thus one has complete control over the amount of energy which is stored in *G* and subsequently utilized for compression of the fresh charge. This enables one to operate with any compression pressure that may be desired. It also enables one to keep the compression pressure constant in *A* and *B*, no matter what may be the pressure of delivery of the water from *K* through the delivery pipe *L*. Thus, if the head is raised, pipe *H* can be raised, and this is the only adjustment necessary to make in pumping against different heads.

At the end of the expansion stroke the gases in the combustion chamber of the Humphrey pump are so cool that fresh mixture may be drawn in and come in contact with the exhaust gases without fear of premature explosion. Fig. 138 shows the combustion chamber of a two-cycle Humphrey pump designed in accordance with the above fact. The combustion chamber has to be specially shaped so that the incoming charge, which may be preceded by pure air, displaces the burnt-products and mixes as little as possible with them. In the sketch *A* is the admission-valve at the top of the tall but narrow part of the chamber *B* in which the full charge volume extends down to the line *cc*. A number of exhaust-valves *E* lead to a common outlet *O* which may be fitted with a non-return valve, or each exhaust-valve may carry a light non-return valve on its spindle as shown. The level at which the expansion reaches atmospheric pressure is, say, *ff*, but this level having been reached by the water its further movement draws in a fresh combustible mixture until this mixture occupies the space down to *cc*, and the liquid level has fallen to *gg*. The column of liquid now returns and drives the exhaust products through the valves *E* (which open by their own weight) until these valves are shut by the water. The kinetic energy acquired by the column is now spent in compressing the fresh charge, which is ignited to start a new cycle. Thus, each out-stroke is a working stroke and no locking gear is required on the valves.

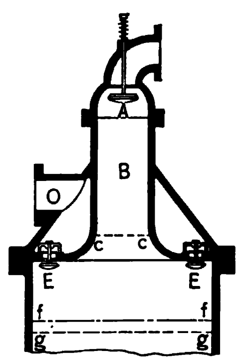


Fig. 138.—Combustion Chamber. Humphrey Two-cycle Pump.

The Humphrey Air Compressor. — If the column of water oscillating in the play pipe of a Humphrey pump is used as a water piston and caused to rise and fall in an air vessel fitted with suitable valves for the inlet and outlet of air, the combination constitutes an air compressor of a very efficient type.

A compressor with a single-barrel pump and a single air cylinder is illustrated in Fig. 139. *A* is the ordinary pump chamber and *C* the air compressor chamber. The inlet-valve *i* for the gas and air and the exhaust-valve *e* exist as before, but the exhaust pipe and valve are adjustable in a vertical direction so as to vary the cushion space at the top of *A*. There is another pipe *p* fitted with a valve *r* and also adjustable vertically. A reservoir *X*, such as a boiler shell, is connected by a flexible pipe to *p*. Corresponding parts are fitted to the compressor; thus, *f* is an

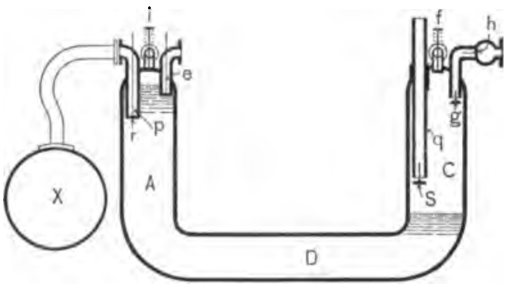


FIG. 139. — Humphrey Simple Air Compressor.

air inlet-valve; *g* an outlet-valve for the air, which is shut by the water; and *h* a non-return valve for the compressed air. The dip-pipe *q* with its valve *S* gives communication between the atmosphere and the compressor. The outlet-pipe and the dip-pipe are adjustable in a vertical direction.

The cycle can be explained by assuming the water level to be high in *A* and low in *C* as shown in the figure, and that a compressed charge is ignited in *A*. Expansion occurs, driving down the water in *A* and up in *C*. At first air escapes from *C* through the valve *S*, but when the water reaches this valve it is shut and the remaining air is trapped and compressed until the pressure of the discharge is reached, when *h* opens and compressed air is delivered.

Next, the water reaches valve *g* and shuts it so that no more air may escape, and the water column is brought to rest by the

continued compression of the remaining air which forms a cushion. Meanwhile in chamber *A* expansion to atmosphere allows the exhaust-valve *e* to open, and the continued stroke draws in scavenging air (the scavenging valve is not shown), but the valves *i* and *r* remain locked. Now the column begins to return, due to the expansion of the cushion air in *C*, and this causes exhaust and cushion in *A* and at the same time there is first a fall of pressure in *C* to atmosphere, then the valve *f* is opened against a light spring, allowing a fresh supply of air to enter.

So far we have had one outward and one return stroke, but the expansion of the cushion in *A* starts a second outstroke which results in drawing in a new combustible charge in *A* through valves *i* and *r*, which were released on the closing of the exhaust valve *e*. As the second outstroke is nearly as long as the first, it follows that too much combustible mixture is drawn in, but the surplus is gotten rid of on the second return stroke by allowing it to escape into the receiver *X*, thus slightly raising the pressure therein. Valve *i* is shut by its spring but *r* is shut by the action of the water, and when this occurs, chamber *A* contains a definite volume of combustible mixture, the size of the charge being fixed by the height of *r*. The second outstroke towards *C* merely compresses the air in *C*, without causing delivery to take place, but stores in the air the energy given out by the expansion of the cushion in *A*. The expansion of the stored air in *C* gives the compression stroke in *A* and the combustible charge is ready to be ignited to start a new cycle.

By making adjustments on the levels of the exhaust-valve *e* and the rejected-charge valve *r*, which are so constructed as to make this possible, and similar adjustments on the compressor side, the machine may be suited to variable pressure and air quantity as desired.

It is also practicable to cause the variation in pressure in *X* to automatically adjust the quantity of water which reciprocates between the chambers to suit the capacity per working cycle, as theory would indicate.

Newer Developments. — One of the newer developments in the Humphrey pump is a device for rendering the compression independent of the head. This device is illustrated in Fig. 140. The chamber at the left represents the combustion chamber of the pump. In this case the water inlet valves are provided near

the delivery end of the pipe as shown at A. The chamber B is an air vessel from which the water may be delivered against a high head. A valve C closes the end of the delivery pipe. On the outstroke this valve is raised by motion of the water, some of which thus escapes into the air chamber. On the return stroke this valve, following the motion of the water, closes the delivery end of the play pipe, and is finally brought quietly to rest by the dash-pot shown at D. The water in the play pipe having, however, already acquired momentum, still moves backward and a fresh supply of water then enters at the inlet-valve. By suitably adjusting the motion of the cut-off valve C any desired proportion of the energy of the cut-off stroke can be utilized in compressing the new charge.

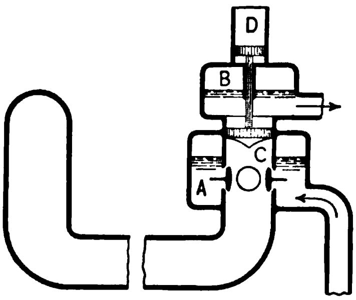


FIG. 140.

A Humphrey pump may also be arranged to have a combustion chamber at each end of the play pipe, as illustrated in Fig. 141,

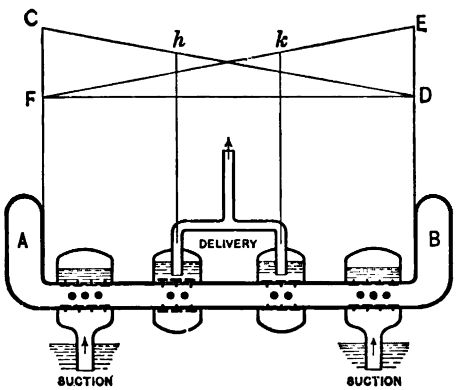


FIG. 141.

where the delivery valves are placed in the region of the pipe where the pressure is most constant. When an explosion occurs in chamber A, and the column of liquid is accelerated, the distribution of pressure along the play pipe is represented by the sloping line CD. At the first delivery box the pressure is represented by h, and so long as this is higher than the back pressure, water

will pass the delivery valves. As the point of delivery approaches nearer and nearer to the combustion chamber, so may the pressure of delivery be increased, and on these lines a pump has been built to pump against a head of 300 feet, including the suction.

A device for increasing the number of cycles per minute has been introduced by Mr. Humphrey and is illustrated in Fig. 142. Referring to Fig. 140, it will be seen that, with the arrangement there illustrated, the water in the play pipe is never changed. Its sole use is by its momentum to provide for the reciprocations required in the combustion chamber. It can therefore be replaced by a mass of metal as indicated in Fig. 142, a device which in certain cases has considerable advantage. In the first place for the same inertia the bulk is reduced to one-seventh of that needed with water, and, secondly, higher rates of retardation and acceleration become possible. With the ordinary type of pump

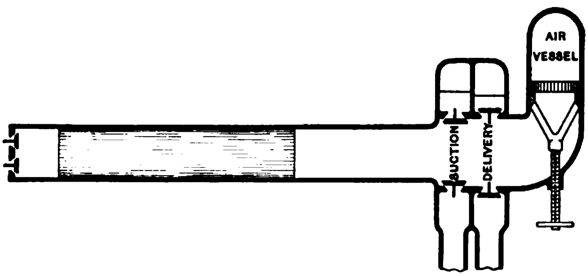


FIG. 142.

it is found that if the retardation on the return stroke is greater than 32 feet per second, water is liable to be sprayed from the surface of the water into the combustion chamber, reducing the thermal efficiency. By using a solid piston this limit to the rate of retardation is eliminated and the rate of reciprocation can be greatly increased. Using a solid piston there is no necessity for making the play pipe horizontal; it may be made vertical without any inconvenience. With an experimental pump of this kind the rate of reciprocation has been increased to 193 cycles per minute.

Efficiency of the Humphrey Pump. — Higher efficiencies than those obtained in the Otto cycle are claimed for the Humphrey pump. Tests made on pumps of this type seem to indicate that the efficiency is actually higher than the efficiency of the reciprocating engine of the same power operating on the same com-

pression. One of the original pumps, a 35 horse-power unit, was tested, and showed efficiencies of 21.25 and 22.07 per cent based on water horse-power. Larger pumps have shown corresponding increase in efficiency.

Advantages of the Humphrey Pump. — The advantages claimed for this pump are as follows:

1. No moving parts required except a few mushroom valves.
2. Practically no lubrication.
3. Simple construction.
4. No cooling jacket.
5. High thermal efficiency.
6. No valves on the delivery side of the pump.
7. No flywheel need be used.
8. Light weight.
9. Low first cost.

CHAPTER XIX

DESIGN OF THE GAS-ENGINE

Calculation of Cylinder Volumes — Allowable Compression — Construction of Indicator Cards — Calculations for Diesel Engines

General. — In any problem of machine design, there is no fixed rule as to the course to pursue; that is, there is no exact method of procedure to follow. There are numerous ways in which a design problem may be worked out, and possibly several of them will appear to be about equal logically. Probably the best course to take is the one involving the fewest assumptions. In any case some assumptions must be made. Some parts must necessarily be designed and redesigned and then designed again in order to fit in with other parts. The beginner in design work does not know just which path will involve the fewest assumptions, and it is only with experience that the faculty for working out a long problem logically can be attained.

The parts of a machine support loads or transmit forces in doing the useful work for which they are designed. They are proportioned to take these loads without breaking and at the same time without an uneconomical expenditure of material. The parts, then, must be strong and rigid. In general, the stronger the material, the less the bulk and consequently the cost. For instance, some bridges have been built of nickel steel instead of carbon steel, even though the nickel steel costs considerably more per pound than the more common kind. The saving was due to the fact that the allowable stress of the nickel steel is much higher than of carbon steel. This reduced the dead weight of the bridge since smaller members could be used. Reducing the dead weight decreased the stresses and permitted a further reduction in the size of the members and consequently in the weight and cost of the bridge.

In starting any design problem, there are four points or milestones, as it were, where halts should be made long enough for

reflection. . These are: first, **analyze**; second, **theorize**; third, **modify**; fourth, **construct.**

The Analysis. — Any problem in design, no matter how small, is important enough to demand careful analysis on the part of the designer. Too little time, rather than too much, is spent on this very important beginning. The success of the whole design rests on the analysis and just as this analysis is accurate, so will the results of the work be of value in proportion. Among the important things to be considered in the analysis are the loads on the part under consideration, how these loads are applied, their magnitudes and directions. The unusual conditions or loads which are likely to come into action during the life of the machine must be taken care of, and, in many cases, it is the unusual that causes trouble.

The Theory. — After the analysis has been made, the conditions that exist must be handled, first, according to theory. Sizes of parts may generally be determined from purely theoretical principles. If several conditions exist at the same time, they must be properly combined by theory. And, as in the analysis, the theory must not be at fault. That is, a bending load must not be treated as a load in straight tension; proper stresses must be used with a given material; loads which the analysis has shown to be suddenly applied must not be treated as dead loads, etc.

Modification. — After the calculations have been made from a theoretical standpoint, they must be changed to suit practical considerations. It is very easy for an engineer to design a machine in the drafting room; it is an entirely different matter to construct this machine in the shop and do it in an economical manner. Parts made of cast-iron require careful consideration in order to make it possible to construct the pattern and mold the piece without excessive cost. Where many parts are to be made identical, they should be modified to allow the use of automatic machinery or jigs. Standard parts should be used when they are available and when they will answer the purpose as well as a specially constructed part.

Construction. — After the modifications have been made, the construction of the drawing may be carried out. During the preliminary and final calculations, free-hand sketches should be made and dimensions attached as they are determined. These

sketches are valuable helps in detecting errors and in making the actual drawing. This drawing should be simple; it should show everything that it is necessary to show and no more. Dimensions should be used freely and should be put on in plain figures. Nothing should be left to the imagination of the mechanic who is to do the actual construction of the part — possibly he will have no imagination.

The Gas-engine Design Problem. — The gas-engine design problem does not differ from any other in that certain conditions must be fixed in order to allow a logical treatment of the subject. We must know what type of engine is to be built, the size and the fuel available. It is also necessary to know, in many cases, the use to which the engine is to be put. We might not design an engine for driving an electric generator just as we would one that is to be belted to a line shaft. The method to be used in this book makes it necessary to know also the back pressure and suction pressure in the cylinder. These two values must be assumed. Further along we will see that other assumptions must also be made.

In order to help the reader in the work that follows, a brief outline is given here of the method of procedure in this problem. The first step is to make calculations that will enable us to find the size of the cylinder. This work has been explained in a previous chapter. After the size of the cylinder has been determined the mean effective pressure may be computed and the indicator card drawn. Since the area of the piston and the pressure on it are known at this point, the load on the engine, that is, the load for which the parts are to be designed, is known, and the design of these parts may begin. The reciprocating parts, piston, rod, crosshead and connecting-rod, are designed first. The weights are found, inertia curves drawn, and these curves combined with the original indicator card giving a curve of net or actual pressures on the piston. These pressures are then transferred to the crank-pin, giving a "rotative effort diagram" from which the weight of the flywheel necessary for the given type of engine is found.

Knowing the weight of the flywheel and the thrust along the piston- and connecting-rods, the shaft may be designed. The cylinder thickness may be determined and the diameter of valves found. After determining the size of the valves, the valve gear-

ing is designed, together with the governor, ignition mechanism and other auxiliaries.

In order to make certain steps and calculations as clear as possible, an example will be worked from time to time to illustrate points in the work as it is developed. For this example we will assume a four-cycle, double-acting engine of 225 brake horse-power, operating at 140 r.p.m. Fuel to be producer gas which has the following composition: H_2 = 14.9 per cent, CO = 19.7 per cent, CO_2 = 10.0 per cent, CH_4 = 3.6 per cent and N_2 = 51.8 per cent, by volume. The suction pressure is assumed to be 13 pounds absolute and the exhaust 16 pounds absolute.

Indicated Horse-power. — The power given in the problem is the output, being smaller than the indicated horse-power by the amount of the friction and auxiliaries. If we assume the work of friction and auxiliaries to be 15 per cent of the total the efficiency will be 0.85. The rated indicated horse-power will then be

$$\frac{\text{rated brake horse-power}}{\text{efficiency}} = \text{rated indicated horse-power.}$$

At this point it is well to find the maximum brake and indicated horse-powers. These may be found by multiplying the rated values by (1 + overload allowance). The figure generally used for overload allowance is 15 per cent. Hence the multiplier becomes 1.15.

TABLE XII

ALLOWABLE COMPRESSIONS FOR VARIOUS FUELS

Fuel.	Compression pressure, lbs. gauge.	Approximate clearance, per cent piston displacement.
Gasoline: Automobile motors, high-speed...	60–90 Aver. 75	32
Stationary slow-speed engines............	70–90 Aver. 80	30
Kerosene: Hot-bulb ignition, fuel injected....	30–75	35–40
Previously vaporized....................	45–85 Aver. 65	35
Natural gas..............................	115	22
Producer gas..............................	100–160	....
Producer gas, water-cooled pistons...........	135	20
Illuminating gas..........................	60–100 Aver. 80	26
Blast-furnace gas..........................	120–190 Aver. 155	17

Suction Displacement per Maximum Indicated Horse-power. — This item has been discussed in Chapters VI and VII but in order to show all the steps we will work out the value for our assumed conditions. First the value of "r," the ratio of volumes at beginning and end of compression, must be determined. We have the pressure at the beginning of compression, 13 pounds absolute. The pressure at the end of compression may be found from Table XII, given on preceding page.

The compression selected for the fuel used in the specimen problem, producer gas, is 140 pounds gauge. The exponent of the compression curve we will assume to be 1.35. The values used by different designers vary from 1.3 to 1.4. The actual value differs with the kind of gas and the efficiency of cooling. The value for adiabatic compression with pure air is 1.406, but with cooling it will fall slightly lower. We then have

$$r = \frac{V_a}{V_b} = \left(\frac{P_b}{P_a}\right)^{\frac{1}{1.35}} = \left(\frac{154.7}{13}\right)^{\frac{1}{1.35}} = \overline{11.9}^{.741} = 6.26.$$

The per cent clearance may be determined at this point, although it is not necessary. V_a represents the volume of gas and air mixture at the beginning of the compression stroke and V_b at the end of the compression stroke. V_a is therefore the volume of the displacement of the piston plus the clearance volume V_b. The per cent clearance is then

$$\frac{V_b}{V_a - V_b} = \frac{1}{\frac{V_a}{V_b} - 1} = \frac{1}{r - 1}.$$

In the problem we have selected, the per cent clearance would be

$$\frac{1}{r-1} = \frac{1}{6.26 - 1} = 0.1904 \quad \text{or} \quad 19.04 \text{ per cent.}$$

It will be seen from the preceding calculations that the value of "r" and hence of the clearance is a function of the suction pressure, compression pressure and exponent of the curve. Fig. 143 shows the relations graphically when $P_a = 13.2$ and 14 pounds per square inch and $n = 1.35$ and 1.30. Fig. 144 gives the same information for suction pressures of 13.2, 14.7 and 16.7 pounds and the exponent of 1.40. The latter figure is sometimes used in oil-engine computations when air alone is compressed.

Before the displacement may be determined,. the thermal efficiency must be assumed. This varies with the compression, being greater for the high compressions. The practical values are found only by actual test, but a table has been prepared showing the efficiencies we might expect for different compression ratios. This is Table V, Chapter VI, and from it we find, by interpolation, that the expected efficiency will be 0.269 for a compression of 140 pounds.

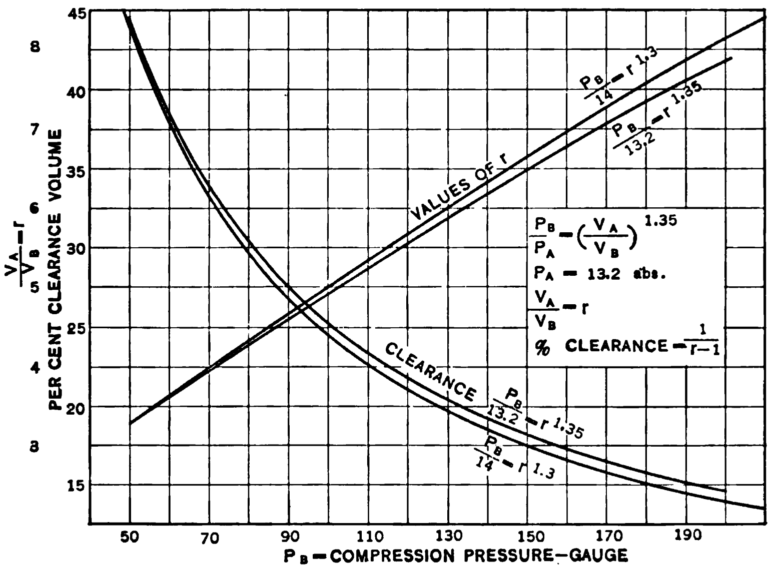


FIG. 143.

The value of $\frac{V_a}{V_0}$ depends on the temperature and pressure, or $\frac{V_a}{V_0} = \frac{P_0 T_a}{P_a T_0}$, where P_0 and T_0 refer to pressure and temperature respectively at standard conditions. All are known except T_a. Here again we must refer to Chapter VI which gives the values 103° F. for water-cooled pistons when $r = 4$, and 89° F. when $r = 7$. Interpolating we find $t_a = 100°$ F. Then

$$\frac{V_a}{V_0} = \frac{14.7}{13.0} \times \frac{560}{522} = 1.21,$$

where 522 is the absolute temperature at 62° F.

The theoretical air required per cubic foot of gas may be found by using Table II. For the gas under consideration this is 1.176 cubic feet. From Table I the heating value is found to be 149 B.t.u. per cubic foot. The suction displacement per minute per maximum indicated horse-power is then

$$S = \frac{\dfrac{42.42}{0.269 \times 149}}{1.21 \times (1.15 \times 1.176 + 1)} = 3.01 \text{ cubic feet.}$$

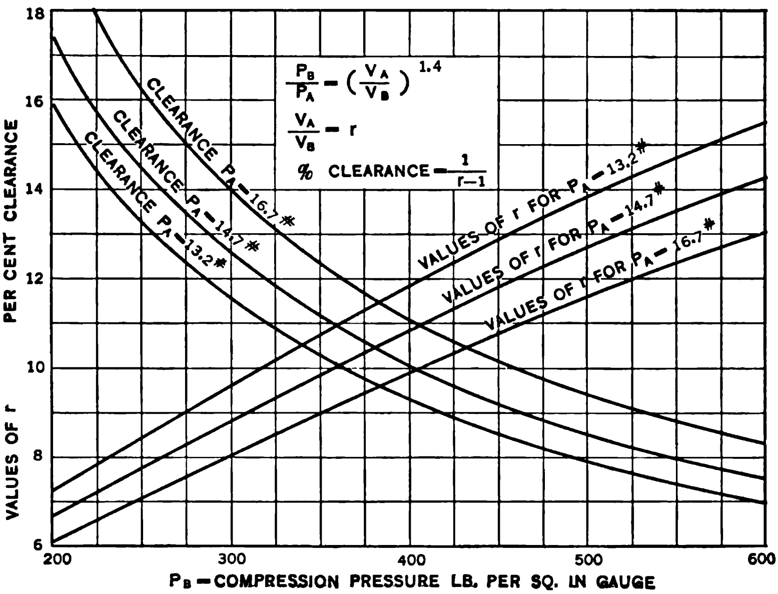


FIG. 144.

From the data given the B.H.P. is 225. Assuming an efficiency of 0.85 and an overload of 0.15 we find the maximum indicated horse-power to be 305. The total volume to be displaced on the suction strokes is then $305 \times 3.01 = 920$ cubic feet. Since the engine is to be four-cycle, double-acting, there will be a suction stroke for each revolution. The volume displaced per minute on the suction strokes will then be $\frac{\pi d^2}{4} \times L \times N$,

where d = diameter of cylinder in feet,
L = length of stroke in feet,
N = r.p.m.

This must be equal to the suction displacement required, 920 cubic feet. Before the equation can be solved, however, it will be necessary to put either L or d in terms of the other.

The ratio of $\frac{L}{d}$ is not fixed by anything except practical considerations. As the lower limit, we might say that it is never made less than one, and for the upper limit we may safely assume two to be the value. The considerations that fix the value between these limits are speed, available space and the shape of the combustion chamber. The tendency in high-speed engines is to make the ratio small, that is, nearer the lower limit than the upper. For a given suction displacement and a given speed there is an infinite number of ratios that could be used. However, the smaller the ratio, the lower will be the piston speed, and it is important to keep down the piston speed in stationary engines.

The space occupied is important in automobile and marine engines. If large values be assumed for this ratio for these types of engines, the result will be high machines of the vertical type that will be top heavy. By making the diameter small and stroke long, the crank radius and connecting-rod length are increased, both helping to increase the height of the unit. This question of long or high units is not so important in stationary engines, and ratios may be made greater in that type.

The shape of combustion chamber is important, particularly in high-compression engines. If the combustion chamber is cylindrical and of the same diameter as the cylinder, the greater the ratio of $\frac{L}{d}$ the longer will be the combustion chamber in proportion to the diameter. If the length of the combustion chamber h is large compared with the diameter, say equal to the diameter, the surrounding area is a minimum in proportion to the volume. The ratio of the enclosed area to the volume of the combustion chamber does not change much until h becomes less than one-third of the cylinder.

In Fig. 145 are shown the different ratios of $\frac{h}{d}$ for two values of $\frac{L}{d}$ and compressions of 110 and 500 pounds gauge. In A the proportion of $\frac{h}{d}$ is not too small to be prohibitive, while that shown

in B is larger than is necessary. A value of $\frac{h}{d}$ between these two, say of $\frac{1}{3}$, would be preferable to either A or B. In C we have the ratio of $\frac{h}{d}$ that would exist in a Diesel engine if $\frac{L}{d} = 1$, and in D the ratio if $\frac{L}{d} = 2$. In both C and D the ratio of $\frac{h}{d}$ is too small for good results. To remedy this, the diameter of the combustion chamber should be made smaller than the cylinder diameter. This is illustrated in Fig. 108.

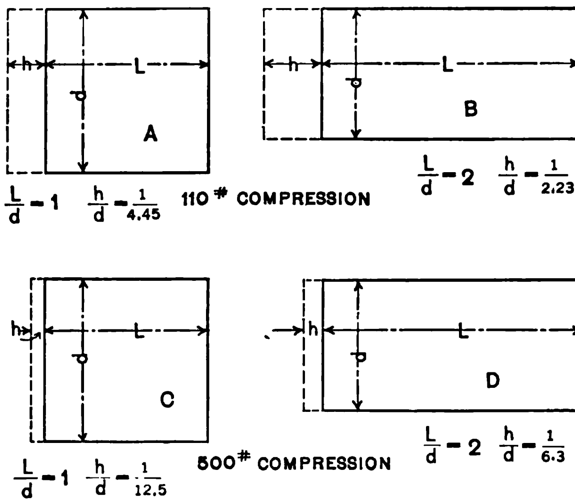


FIG. 145.

From the foregoing it will be seen that a value of $\frac{L}{d}$ of 1.5 will give good proportions in the combustion chamber when the compression is in the neighborhood of 140 pounds. Using this ratio for $\frac{L}{d}$ we have the equation

$$\frac{\pi d^2}{4} \times 1.5\, d \times N = 920 \text{ cubic feet.}$$

Then $$d = \sqrt[3]{\frac{920 \times 4}{\pi \times 1.5 \times 140}} = 1.75 \text{ feet} = 21 \text{ inches.}$$

The stroke will then be $1.5 \times 21 = 31.5$ inches.

The Indicator Card. — This card should be drawn next but the pressure at the end of combustion has not yet been determined.

One method of determining this pressure was given in Chapter VIII but as this method is laborious and of doubtful value, another method will be given here.

The mean effective pressure may be found by the equation

$$\text{I.H.P.} = \frac{\text{m.e.p.} \times L \times A \times N}{33{,}000},$$

then $$\text{m.e.p.} = \frac{\text{I.H.P.} \times 33{,}000}{L \times A \times N}.$$

The problem then is to draw a card to give a mean effective pressure as above. The points A and B, Fig. 146, have already been found. The point x, the starting point of the expansion line, is to be determined. The area under xD is $\dfrac{P_xV_x - P_dV_d}{n-1}$ and under the compression line AB is $\dfrac{P_aV_a - P_bV_b}{n-1}$. The difference of these two areas will be the work represented by the indicator card and if this area is divided by the length of the card $V_a - V_b$ the result will be the m.e.p. But to allow for rounding the corners of the card we must increase the m.e.p. by using a diagram factor. If we used the m.e.p. as found to construct the card $ABxD$ with square corners, the card will be too small after the corners are rounded as in the actual card. The diagram factor f will be from 0.96 to 0.98. Then

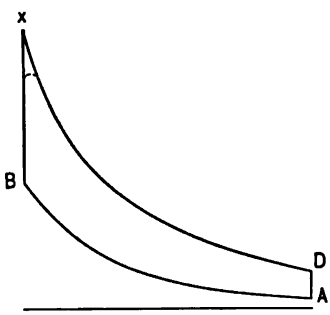


Fig. 146.

$$\frac{\text{m.e.p.}}{f} = \frac{\dfrac{P_xV_x - P_dV_d}{n-1} - \dfrac{P_bV_b - P_aV_a}{n-1}}{V_a - V_b},$$

m.e.p. = pounds per square inch.

P_a, P_x, etc., are pounds per square inch absolute.

Since combustion takes place at constant volume $V_b = V_x$ and for convenience it may be assumed that $V_x = V_b =$ unity.

Then $$V_a = V_d = r, \quad V_a - V_b = r - 1.$$

Hence
$$\frac{\text{m.e.p.}}{f} = \frac{\dfrac{P_xV_x - \dfrac{P_xV_x^n}{V_d^{n-1}}}{n-1} - \dfrac{P_bV_b - P_aV_a}{n-1}}{V_a - V_b}.$$

This equation contains only one unknown P_x which may be solved for. The expansion line xD should be started from P_x and the compression line from P_b. The point of the card should be rounded off below P_x, say at 350 to 450 pounds, depending on the kind of fuel.

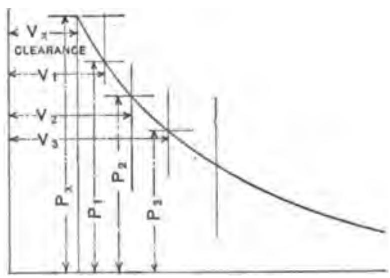


FIG. 147.

The expansion and compression curves may be plotted as follows:

$$P_xV_x^n = P_1V_1^n,$$

$$P_1 = P_x\left(\frac{V_x}{V_1}\right)^n. \qquad \text{Let} \quad \left(\frac{V_x}{V_1}\right)^n = K.$$

Let $\dfrac{V_x}{V_1}$ = constant = C. That is, select ordinates through 1, 2, 3,

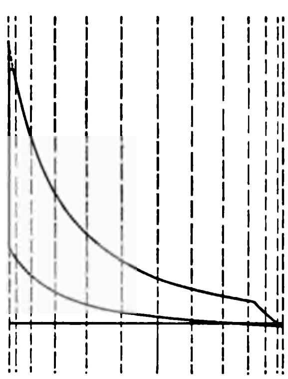


FIG. 147a.

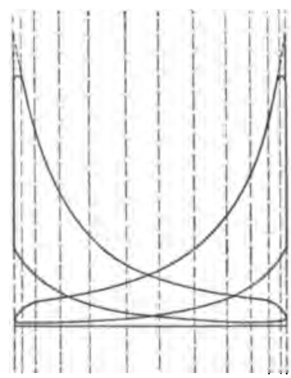


FIG. 147b.

etc., so that the volume at each will be the volume at the preceding ordinate divided by constant C. Thus $V_1 = \dfrac{V_x}{C}$, $V_2 = \dfrac{V_1}{C}$, etc. Then the successive pressures may be calculated and plotted. $P_1 = KP_x$, $P_2 = KP_1$, etc., where $K = C^n$. By this method,

illustrated in Fig. 147, it is necessary to raise only one number C to the nth power. The value of n may be assumed to be 1.35. The completed indicator cards should have the appearance of those shown in Figs. 147a and 147b, which represent cards from single- and double-acting engines respectively.

Oil-engine Design. — The design of a Diesel or semi-Diesel engine should be started with an assumed value of the thermal efficiency. From this the oil required per I.H.P. hour may be found if the heating value of the oil is known. The next quantity to find is cubic feet of air required for the amount of oil that is to be delivered to the engine for each explosion. In finding this amount of air it will be necessary to use the temperature and pressure in the cylinder at the end of the suction stroke and reduce the volume to correspond to that condition. Allowance should be made for 50 per cent excess of air and from 10 to 15 per cent overload.

After the volume of air required per suction stroke is determined, the diameter and stroke may be found thus:

$$\frac{\pi d^2}{4} \times L = \text{air required per stroke},$$

d and L being diameter and length of stroke in feet.

To illustrate this calculation, let us find the cylinder dimensions for a 50 B.H.P. four-cycle, single-cylinder, single-acting, 225 r.p.m. Diesel engine. Assuming a mechanical efficiency of 80 per cent and an overload of 15 per cent, the maximum indicated horse-power will be

$$\frac{50}{0.80} \times 1.15 = 72.$$

If the fuel oil used be assumed to have a composition of $H_2 = 14$ per cent, $C = 84$ per cent, Balance $= 2$ per cent, by weight, we find the air required per pound to be, Table II,

Carbon............	$152 \times 0.84 =$	128
Hydrogen.........	$456 \times 0.14 =$	64
Total		192 cubic feet at 62° F.

Allowing 50 per cent excess, total is $192 \times 1.5 = 288$ cubic feet. Assuming suction pressure to be 13.2 pounds per square inch and

temperature 120° the total volume of air after it is drawn into the cylinder will be

$$\frac{288 \times 14.7 \times 580}{13.2 \times 522} = 357 \text{ cubic feet.}$$

In order to find the heating value of the oil we will use the equation given in Chapter V,

$$\text{B.t.u.} = 14{,}500\,C + 52{,}230\left(H - \frac{O}{8}\right)$$
$$= 14{,}500 \times 0.84 + 52{,}230 \times 0.14 = 19{,}480.$$

If an efficiency of 0.35 based on maximum I.H.P. is assumed, the fuel consumption per maximum I.H.P. hour will be

$$\frac{2545}{0.35 \times 19{,}480} = 0.373 \text{ pound.}$$

The volume of air to be drawn in at each suction stroke will be

$$\frac{0.373 \times 72 \times 2 \times 357}{60 \times 225} = 1.42 \text{ cubic feet,}$$

which is the piston displacement.

If we now assume a ratio of stroke to diameter of 1.5 to 1, we have

$$\frac{\pi d^2}{4} \times 1.5\,d = 1.42,$$

$$d = \sqrt[3]{\frac{4 \times 1.46}{\pi \times 1.5}} = 1.073 \text{ feet} \quad \text{or} \quad 13 \text{ inches.}$$

The stroke will be $1.5 \times 13 = 19.5$ inches.

The horse-power formula for this type of engine is

$$\text{I.H.P.} = \frac{PLAN}{2 \times 33{,}000},$$

where N is the r.p.m. If we use this formula to solve for the maximum m.e.p., we must put in the maximum indicated horse-power, 72 in this case. Then

$$\text{m.e.p.} = \frac{2 \times 33{,}000 \times 72}{\frac{19.5}{12} \times 133 \times 225} = 98.5 \text{ pounds per square inch.}$$

The Indicator Card. — In constructing the indicator card for the Diesel engine, it will be necessary to determine the point of cut-off, or the point V_x, Fig. 148. Proceeding in a manner similar

to that for the Otto card, we find the following equation to be true:

$$\frac{\text{m.e.p.}}{\text{diag. factor}} = \frac{P_b(V_x - V_b) + \dfrac{P_xV_x - P_dV_d}{n-1} - \dfrac{P_bV_b - P_aV_a}{n-1}}{V_a - V_b}$$

$$= \frac{P_b(V_x - V_b) + \dfrac{P_bV_x - \dfrac{P_bV_x^n}{V_d^{n-1}}}{n-1} - \dfrac{P_bV_b - P_aV_a}{n-1}}{V_a - V_b},$$

where m.e.p. is pounds per square inch as already found and P_a, P_b, etc., are pounds per square inch absolute.

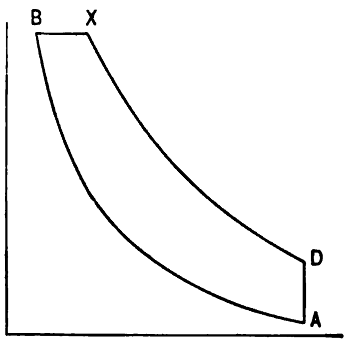


FIG. 148.

Assume for convenience that

$$V_b = 1.$$

Then
$$V_a = V_d = r,$$
$$V_a - V_b = r - 1,$$
$$n = 1.35,$$

Diagram factor = 0.96 to 0.98.

Theoretically the value of n for the compression of pure air is not the same as the n for the expansion of the burned gases. Actually they are nearly the same and lie between 1.30 and 1.35, depending on the type of engine.

The value of $P_b = P_x$ is usually 500 pounds for a Diesel engine. The pressure at the beginning of compression P_a will be atmospheric or 2 or 3 pounds above for a two-cycle engine, and will be 1.5 or 2 pounds below the atmosphere for a four-cycle engine. The value of the clearance should be found as for the Otto cycle engine.

As a matter of convenience, it will be well to make the indicator card about 4 inches long, and some even proportion of the stroke. Thus for a 16-inch stroke the card would be made one-fourth the stroke, and for a 24-inch stroke it would be made one-sixth the stroke. The spring (vertical) scale used in drawing the indicator cards may be 100 pounds to the inch for all types of engines. The base-scale and the area-scale should be computed, the former being the number of feet on the stroke represented by each inch of length on the indicator card. The area-scale is the

product of the spring- and base-scales and the unit is foot-pounds per square inch of indicator card.

The mean effective pressure of the finished card should be measured by a planimeter. The point of release must be assumed, however, before the card may be completed. The release should occur with the crank 30 to 40 degrees before dead center for a four-cycle engine. Release occurs on a two-cycle engine when the piston uncovers the exhaust-ports, the ports being made to give proper exhaust velocities. This will be taken up in a later chapter.

In a four-cycle engine release may be obtained on the card as shown in Fig. 149. The crank semi-circle should be laid off, as shown, the proper distance from the card. Draw in crank position as shown, 30 to 40 degrees before dead center and with a

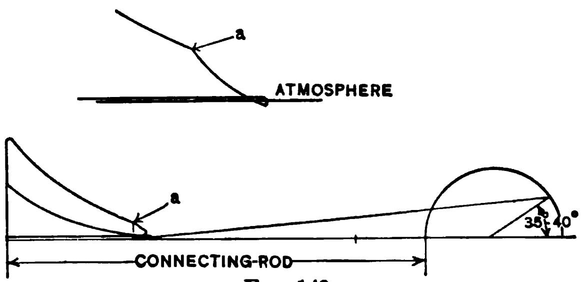


FIG. 149.

radius equal to the length of the connecting-rod, find the point of release on the stroke (card base) and project it up to the expansion line as at *a*, Fig. 149. From *a* to the end of the stroke the curve will be free expansion and will have the form shown in the large scale detail. It will, in some cases, go down below the atmosphere, particularly in engines with long exhaust pipes. In drawing the card with a spring scale of 100 pounds, it will be sufficient to carry this line down to the atmospheric line.

EXERCISES

1. Find the air required per cubic foot of gas of the following composition by volume:

	Per cent
H_2	= 32
CO	= 43
O	= 2
CH_4	= 3
CO_2	= 6
N	= Bal.

2. (*a*) Find heating value per cubic foot at 32° F. of gas given in Prob. 1. (*b*) Find heating value per cubic foot of suction displacement if temperature of charge in cylinder is 120 degrees, pressure 13.4 lbs. per sq. in. abs.

3. (*a*) Find suction displacement per maximum I.H.P. per minute with gas given in (1) and conditions given in (2). Efficiency 23 per cent. (*b*) Find size of cylinder to develop 125 horse-power, $\frac{L}{d} = 1.5$, r.p.m. = 175, engine to be two-cycle, single-cylinder, single-acting.

4. Find the pressure p_x for the starting point of the expansion line of an indicator card for an m.e.p. of 84 lbs., $V_b = 1$, $V_a = 6.5$, $P_b = 140$ lbs. gauge, $P_a = 13.7$ lbs. abs. Diagram factor 92 per cent. Assume $n = 1.35$.

5. (*a*) Find the cylinder dimensions for a Diesel engine of 75 horse-power, single-acting, four-cycle, single-cylinder, 200 r.p.m. Fuel oil composition, C = 84 per cent, H_2 = 15 per cent. Excess air = 50 per cent. Efficiency based on I.H.P. = 40 per cent. Suction pressure 14 lbs. abs., temperature 115° F. $\frac{L}{d} = 125$. (*b*) Find the mean effective pressure of the engine.

CHAPTER XX

DESIGN OF THE GAS-ENGINE (Continued)

Piston — Piston-rings — Wrist-pin — Piston-rod — Crosshead — Connecting-rod

The Trunk Piston. — In a single-acting engine a **trunk piston** is usually used and more often than not it is used without a crosshead; that is, the piston takes the vertical component S of the connecting-rod thrust instead of a crosshead taking it. The bearing area between the trunk piston and cylinder wall is based on the maximum vertical component which should be found graphically, by trial. This maximum occurs usually between 30 and 40 degrees crank angle.

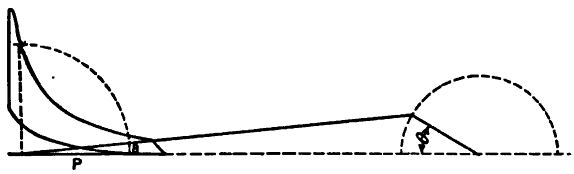


FIG. 150.

In order to allow for rings in the piston the total bearing area should be considered to be the product of 0.8 of its length by 0.85 of its diameter. The bearing area is then 0.68 DL', where D is the diameter and L' the length of the piston. The bearing pressure should not exceed 30 pounds per square inch of this net area, and where a long piston may be used without increasing the weight of reciprocating parts to a prohibitive figure, the pressure should be 20 pounds. Thus for fast running engines the higher value should be used and for slower engines the lower. In no case should the length of piston be less than the diameter. In **stationary** engines it is better to have length of piston 1¼ to 1½ times the diameter.

In Fig. 150 is shown the construction for finding maximum value of S. This should be done for several values of θ, the value of P being taken from the indicator card corresponding to the

value of θ. After S_{max} is determined, the length of the piston may be found. $L' = \dfrac{S \times \text{Area piston}}{0.68\,Dp}$, where S = vertical component pounds per square inch of piston, L' = length of piston in inches, D = diameter piston in inches, and p = allowable bearing pressure in pounds per square inch of projected area.

Strength of Piston. — The **thickness** of the **face** of the piston may be found by considering it a flat round plate supported at the edges. The dangerous section of such a plate is along a diameter. This section is under the action of two moments acting in opposite directions. The first is due to the fluid pressure $\dfrac{\pi D^2}{4 \times 2} p_{max} = \dfrac{\pi R^2}{2} p_{max}$ acting at an arm $\dfrac{4}{3}\dfrac{R}{\pi}$ so that the moment is $\frac{2}{3} R^3 p_{max}$. The second results from the reaction at the edge, which is $\dfrac{\pi R^2}{2} p_{max}$ acting at a lever of $\dfrac{2\,R}{\pi}$, giving the moment $R^3 p_{max}$.

The resulting moment is $R^3 p_{max} - \frac{2}{3} R^3 p_{max} = \frac{1}{3} R^3 p_{max}$. (1)

The section modulus of a flat plate of a thickness t and diameter D is

$$\frac{Dt^2}{6} = \frac{Rt^2}{3}. \tag{2}$$

Equating the resisting moment to the bending moment we have

$$K_b \frac{Rt^2}{3} = \frac{R^3 p_{max}}{3}, \tag{3}$$

$$t = R\sqrt{\frac{p_{max}}{K_b}}, \tag{4}$$

where K_b is the allowable stress in bending.

Bach writes this formula

$$t \geqq r\sqrt{n\frac{p_{max}}{K_b}},$$

where $n = \frac{5}{6}$ for supported plates and $\frac{2}{3}$ for fixed plates and r is the **inner** radius. In formula (4) we have assumed R to be $\dfrac{D}{2}$, the outer radius, and have assumed the plate to be supported instead of fixed. These assumptions make (4) extremely safe so that a high stress may be used, say 8500 pounds per square inch.

To make the piston stronger still, the head is often ribbed

inside with four or six ribs of a thickness one-third to one-half the thickness of the face. These ribs are radial, and may extend back to the open end of the trunk or may stop at the piston-pin boss.

The maximum thickness of the piston **barrel** may be found by the formula

$$t_1 = \left(\frac{D}{50} + t_r + 0.2\right) \text{inches},$$

where D = diameter cylinder in inches,
t_r = thickness ring in inches,
t_1 = thickness barrel.

The decrease of t_1 toward the open end depends on the webbing and on the considerations of manufacture. The thickness at end may be $\frac{1}{2}\,t_1$ to $\frac{1}{3}\,t_1$. To allow for expansion due to high temperatures the piston barrel should be tapered from the last ring-groove to the closed end. The amount of this taper varies with the size, kind of service and with the material. It may be found correctly by experience only, but is usually 0.2 to 0.5 per cent of the diameter.

The first **ring-groove** should be at a distance from the end of the piston at least equal to the piston-face thickness, better 1.3 times the face thickness. The thickness of the bridge between the grooves should be at least equal to the width of the ring. The depth of the ring-groove should exceed the thickness of the ring by not more than 0.02 inch. If a greater clearance is allowed, it fills up with oil and the oil thickens, sticking the ring fast in the groove.

To obtain uniform distribution of thrust between piston and cylinder wall, the **wrist-** or **piston-pin** should be placed in the center of the part of piston carrying the load. If the combined width of rings be subtracted from the total piston length, half the remainder will be the proper distance for the center of pin from the open end.

Piston-rings. — These rings are usually plain cast-iron snap rings of small width, fitted in grooves cut around the circumference of the piston. Special spring rings and detachable seats for the rings have not proven successful, as nearly all these special constructions lose their efficiency after a short time by burning or sticking fast. In general the snap rings are made of **uniform thickness** as **eccentric** rings do not present any great

advantage. On the other hand they cost more to make and toward the joint the clearance at the bottom of the groove is increased to a dangerous degree.

In Fig. 151 is shown a piston ring sprung to a diameter D, the bore of the cylinder. The pressure per square inch between ring and cylinder wall is p pounds. If we assume the width of

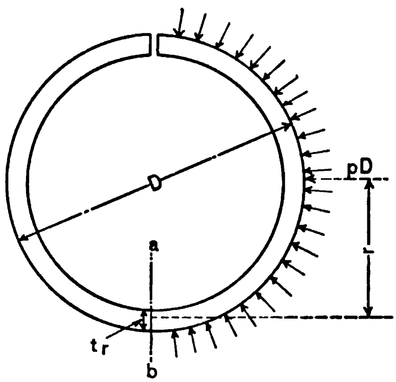


Fig. 151.

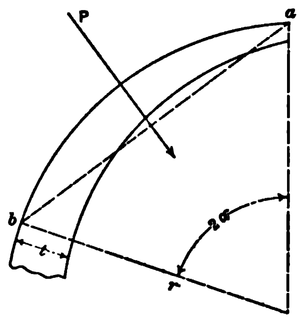


Fig. 152.

ring to be unity, then the pressure on half the circumference of the ring will be pD and this pressure will be exerted as indicated by pD in the figure. The moment of this force about the plane ab opposite the cut in ring is approximately $p\,Dr = 2\,pr^2$ where r is the radius of cylinder in inches.

The resisting moment of ring is $\frac{K_b t_r^2}{6}$, where t_r = thickness of ring in inches.

Then
$$2\,pr^2 = \frac{K_b t_r^2}{6}, \tag{1}$$

or
$$\frac{r}{t_r} = \sqrt{\frac{K_b}{12\,p}}. \tag{2}$$

From this equation the allowable thickness of the ring may be found. The allowable stress may be taken very high, 12,000 to 17,000 pounds per square inch for cast-iron.

Eccentric Rings. — In Fig. 152 is shown a short section of a ring of varying thickness, initially circular on the outside, which has been sprung into a cylinder of smaller diameter. Let I be the moment of inertia of the section at b, which may be taken

normal to the outside curve of the ring. Let ρ be the initial outside radius of the ring and t the thickness at b, all dimensions in inches. It will be necessary to make the ring of uniform width w. It is required to find the varying thickness t which will produce a uniform pressure of p pounds per square inch between the ring and cylinder walls.

The pressure of the ring on the cylinder per unit of length is pw, and the result of the uniform pressure from a to b is $pw \times$ chord ab or

$$P = 2\,pwr \sin \alpha. \tag{1}$$

The moment of this about b is

$$2\,pwr^2 \sin^2 \alpha. \tag{2}$$

Putting this value in the equation for the bending moment,

$$M = EI\left(\frac{1}{r} - \frac{1}{\rho}\right), \tag{3}$$

where $$I = \frac{wt^3}{12},$$

hence $$\frac{1}{r} - \frac{1}{\rho} = \frac{24\,pr^2 \sin^2 \alpha}{Et^3}. \tag{4}$$

For $2\,\alpha = 180$ degrees let t_r be the thickness of the ring.

Then $$\frac{1}{r} - \frac{1}{\rho} = \frac{24\,pr^2}{Et_r^3}. \tag{5}$$

Combining (4) and (5)

$$\frac{t}{t_r} = \sqrt[3]{\sin^2 \alpha}. \tag{6}$$

A table of values for t for different angles is given here:

$2\,\alpha$	t
10	0.197 t_r
20	0.311
40	0.489
60	0.630
80	0.745
100	0.837
120	0.908
140	0.960
160	0.990
180	1.000

Fig. 153 shows a ring drawn from values given in the above table. About two-thirds of the inside curve agrees closely with a circular arc struck from the point C at a distance a from the center of the outside ring. It may be shown graphically that $a = 0.21\ t_r$,

approximately. The ring can be bored to a radius $r - 0.79\,t_r$ and the points near A thinned to agree with the proportions above.

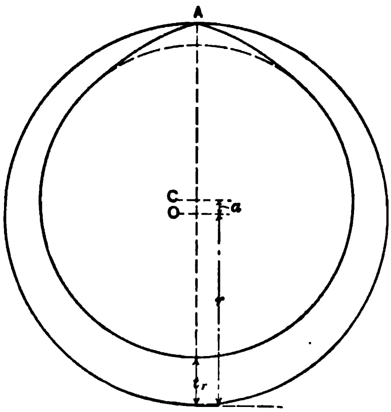


FIG. 153.

From equation (5)

$$p = \frac{Et_r^3}{24\,r^2}\left(\frac{1}{r} - \frac{1}{\rho}\right). \qquad (7)$$

The stress in the ring at its thickest part is

$$K = \frac{Md_1}{I} = \frac{6\,M}{bt_r^2}, \qquad (8)$$

$$M = EI\left(\frac{1}{r} - \frac{1}{\rho}\right) = \frac{Ebt_r^3}{12}\left(\frac{1}{r} - \frac{1}{\rho}\right).$$

Hence
$$K = \frac{Et_r}{2}\left(\frac{1}{r} - \frac{1}{\rho}\right). \qquad (9)$$

Using this value in (7)

$$K = 12\frac{pr^2}{t_r^2}, \qquad (10)$$

$$t_r = r\sqrt{\frac{12\,p}{K}} \qquad (11)$$

which is the same equation derived for rings of uniform thickness in the last article.

From equation (9) we get

$$\rho = \frac{rEt_r}{Et_r - 2\,Kr}, \qquad (12)$$

which gives us a value for the outside radius of the ring before it is sprung into the cylinder. The piece to be cut out is $2\,\pi\rho - 2\,\pi r$ and may be found from (12) as follows:

$$2\,\pi\rho - 2\,\pi r = \frac{4\,\pi Kr^2}{Et_r - 2\,Kr}. \qquad (13)$$

The stress due to slipping the ring over the piston to get it in the groove may be greater than that due to springing the ring into the cylinder. In slipping the ring over the end of the piston the outside radius of the ring is increased from ρ to R where $R = r + t_r$.

Then

$$\frac{1}{\rho} - \frac{1}{r + t_r} = \frac{2K'}{Et_r},$$

$$2K' = \frac{Et_r}{\rho} - \frac{Et_r}{r + t_r},$$

$$K' = \frac{Et_r(r + t_r - \rho)}{2\rho(r + t_r)}. \qquad (14)$$

K' should not be greater than 17,000. In some cases it will be less than K, the stress due to springing the ring into the cylinder; in other cases it will be greater. Usually K' increases with increasing values of p, the unit pressure between the ring and the cylinder walls.

In order to show the application of the above formula, the rings for a 21-inch diameter cylinder will be calculated here. Assuming a pressure p of 4 pounds per square inch, a stress of 15,000 pounds per square inch, and E = 17,000,000 we find from (11)

$$t_r = 10.5\sqrt{\frac{12 \times 4}{15{,}000}} = 10.5 \times 5.657 = 0.59 \text{ inch.}$$

From (12) we find the value of

$$\rho = \frac{10.5 \times 17{,}000{,}000 \times 0.59}{17{,}000{,}000 \times 0.59 - 2 \times 15{,}000 \times 10.5} = 10.84 \text{ inches.}$$

The part cut out will be 2π (10.84 − 10.50) = 2.14 inches. The stress due to slipping the ring over the end of the piston may be found from (14),

$$K' = \frac{17{,}000{,}000 \times 0.59\,(10.5 + 0.59 - 10.84)}{2 \times 10.84\,(10.5 + 0.59)} = 10{,}430 \text{ pounds.}$$

If, instead of using 4 pounds per square inch of projected area between the piston rings and cylinder walls, we use 5 pounds, keeping the other values the same, we find the thickness of the ring = 0.66 inch, ρ = 10.80 inches, the part cut out = 1.88 inches and the stress K' = 16,750.

As stated before, the width of the ring has no influence on the stress, so that this dimension may be assumed arbitrarily. It may be assumed to be from $1\frac{1}{4}$ to 2 times the thickness t_r, depending on the **number** of rings used. The number of rings may be found by

$$n = \frac{D}{5w},$$

where D is the cylinder diameter and w the width of one ring, both in inches. It is of practical advantage to use narrow rings in sufficient number rather than wider rings, fewer in number.

Piston-pin. — The piston-pin should be designed to carry the maximum total load on the piston, that is, the load due to explosion pressure. The pin may be proportioned so the ratio of $\frac{l}{d} = \frac{4}{3}$, where l = length and d = diameter of pin. The **projected area** will then be ld. Assuming an allowable bearing pressure of 1700 to 2000, p', pounds per square inch of projected area, we have

$$p'ld = p_{max}A, \tag{1}$$

where p_{max} = explosion pressure from card,

A = area cylinder in square inches.

If we use the ratio $\frac{l}{d} = \frac{4}{3}$, then

$$\tfrac{4}{3} \times p'd^2 = p_{max}A, \tag{2}$$

which may be solved for d. Then $l = \frac{4}{3}\, d$, which should give a value $< \frac{D}{2}$ for convenient construction.

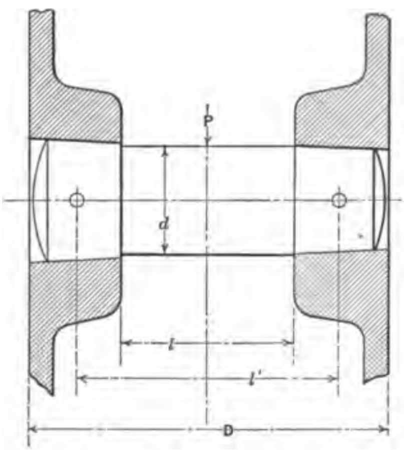


Fig. 154. — Piston-pin.

Strength of Pin. — The pin is fitted in the piston as shown in Fig. 154. We may assume it to be uniformly loaded as a beam for the distance l, with supports at the centers of the bosses at either end, making the length of beam between supports l'. Since l has already been determined, l' may be found from the relation

$$l' = l + \frac{D - l}{2} = \frac{l + D}{2}. \tag{3}$$

Since the load $p_{max} \times$ area $= P$ is evenly distributed over the length l, the maximum bending moment will be at the center of the pin, or

$$M = \frac{P}{2}\left(\frac{l'}{2} - \frac{l}{4}\right). \tag{4}$$

This should be equated to the resisting moment of the pin in bending $K_t \frac{\pi d^3}{32}$, and the stress K_t solved for. This should fall below 12,000 pounds per square inch.

For the 21-inch cylinder of the specimen problem, assuming an explosion pressure of 425 pounds, we would have

$$\tfrac{4}{3} \times 2000\, d^2 = 425 \times 346,$$

$$d = \sqrt{\frac{3 \times 425 \times 346}{4 \times 2000}} = 7.43 \text{ or } 7\tfrac{7}{16} \text{ inches},$$

$$l = \tfrac{4}{3} \times 7.43 = 9.92 \text{ or } 10 \text{ inches},$$

$$l' = \frac{10 + 21}{2} = 15.5 \text{ inches},$$

$$M = \frac{425 \times 346}{2} \left(\frac{15.5}{2} - \frac{10}{4}\right) = 386{,}000 \text{ inch-pounds},$$

$$K_t = \frac{32 \times 386{,}000}{\pi \times \overline{7.43}^3} = 9410 \text{ pounds per square inch.}$$

The **piston-pin** may be held in place in the piston in any one of several ways. Some of the different ways are illustrated in Figs. 155 to 158. In Fig. 155 are shown two **set-screws,** with locknuts. This is an inexpensive and convenient method of holding the pin in small engines with a solid-end connecting-rod. Where the connecting-rod has a wedge adjustment, the set-screws interfere with the use of wrenches to take up such adjustment. The diameter of the set-screw may be $0.2\, d + \tfrac{3}{16}$ inch, where d = diameter of piston-pin.

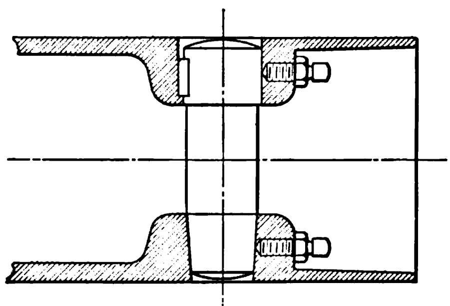


FIG. 155.

The **taper pin** shown in Fig. 156 may be used in preference to the set-screws when set-screws interfere with adjusting the rod. The diameter of the pin may be $0.15\, d + \tfrac{3}{16}$ inch. The taper pin and the set-screws are both unsatisfactory for fixing the piston-pin in large pistons. Much better ways are shown in Figs. 157 and 158. The method used in Fig. 157 is liable to spring the

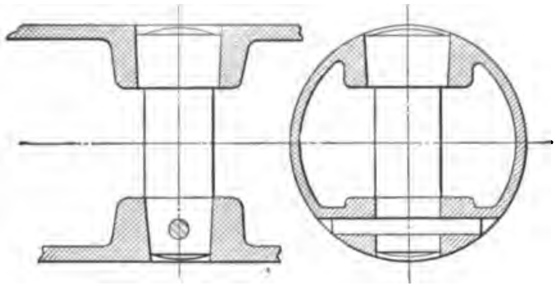


FIG. 156.

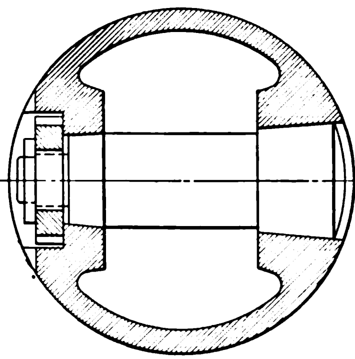


FIG. 157.

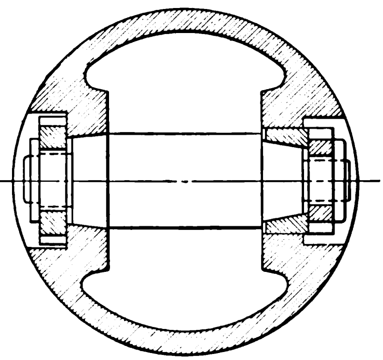


FIG. 158.

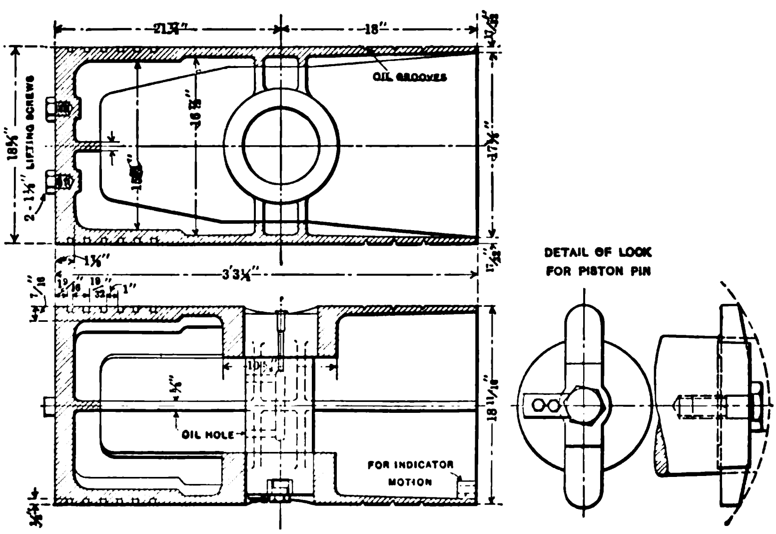


FIG. 159. — Typical Gas-engine Piston.

piston in large engines so that the method shown in Fig. 158 is often used.

A typical gas-engine **piston** is shown in Fig. 159 for a cylinder $18\frac{5}{8}$ inches diameter. The piston is ribbed and has oil-grooves near

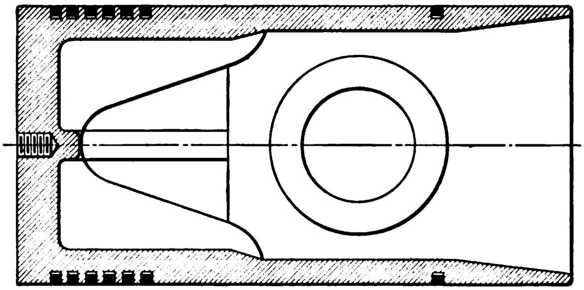


Fig. 160. — Diesel Engine Piston.

the open end instead of a snap ring used by many manufacturers. The pin is retained as shown in detail at the right of the figure. This method is not good for small engines.

The diameter of the boss may be made $1\frac{1}{2}\,d + 1$ inch, where d is the diameter of the pin.

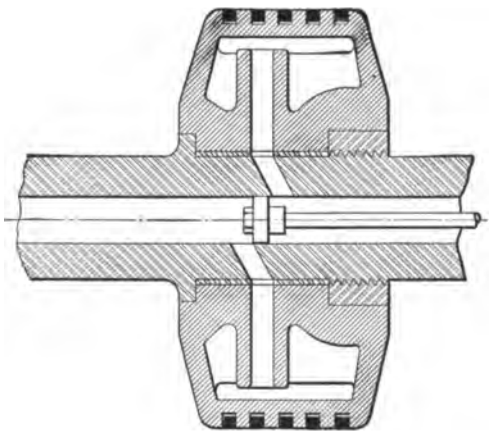


Fig. 161.

A **Diesel** engine piston is shown in Fig. 160. Here a cast-iron ring is used to retain the oil in place of the grooves shown in Fig. 159. This piston illustrates effectively the massive construction necessary in an engine of the Diesel type where pressures in starting are just about double the pressures found in a gas-engine.

Double-acting pistons are illustrated in Figs. 161 and 162. This type of piston is always cooled with water as there is no chance whatever for radiation or conduction to the atmosphere direct. The water is conducted to the piston by means of the hollow piston-rod. Water enters the **bottom** and leaves at the top of the piston. This arrangement requires that there shall be a stop in the hole in the rod between the inlet and outlet openings at the piston. In Fig. 161 this stop is a separate piece held in place by a rod extending to the crosshead. In Fig. 162 the stop is formed by metal left at that point when the hole is drilled in the rod.

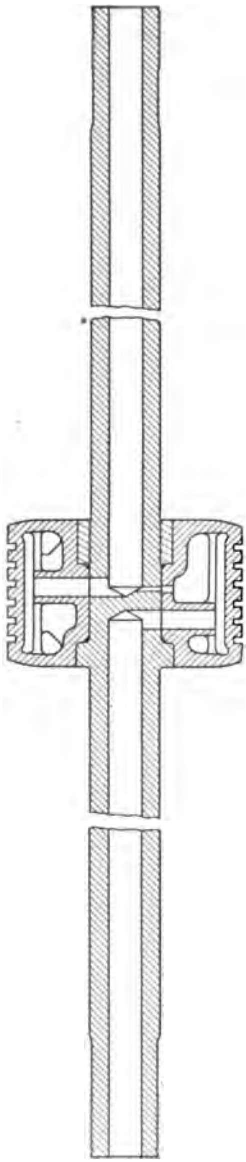


Fig. 162.

The **thickness** of the double-acting piston may be a little less than the thickness of the head of a trunk piston. Care must be taken to avoid straight lines and sharp corners. In other words, the casting must be as flexible as it is possible to make it. The piston is fixed to the rod by means of a nut which forces the piston against a shoulder on the rod. This nut is made round where it enters the piston, with a hexagonal head. After the nut has been tightened up as far as possible, the head is turned off flush with the piston. The collar on the rod may be designed for shear, but in such case a low fibre stress should be used. The diameter of the collar may be $1\frac{1}{4}\,d + \frac{1}{4}$ inch, where d is the rod diameter. The rod must be smaller on one end than the other in order to allow the threads to pass through the piston. The **threads** are usually made fine, say 8 to the inch, and of U. S. Standard form.

Piston-rod Design. — The **piston-rod** of a gas-engine is subjected to alternate tension and compression and as the stresses alternate every stroke, the repetitions are frequent enough to make the application of the principles of fatigue a necessity. For this reason a low fibre stress should be used. The **load** on the rod is equal to the area of the piston multiplied by the explosion pressure.

$$\frac{\pi D^2}{4} \times p = K_t \frac{\pi d^2}{4}. \tag{1}$$

$$d = D\sqrt{\frac{p}{K_t}}. \tag{2}$$

In our specimen problem the diameter of the cylinder was found to be 21 inches, and explosion pressure of 425 pounds may be assumed for convenience. Then

$$d = 21\sqrt{\frac{425}{6000}} = 5\tfrac{3}{4} \text{ inches.}$$

This diameter is for a **solid rod.** To find the diameter of a **hollow rod** to correspond, it is only necessary to assume the diameter of the hole and add the area of the hole to the area of a $5\frac{3}{4}$-inch circle. The diameter corresponding to the sum of these areas will be the outside diameter of the hollow rod. The hole in the rod should be from 2 to 6 inches, depending on the size of the piston. In this case if we assume a 3-inch hole, the necessary outside diameter of the hollow rod will be $6\frac{1}{2}$ inches.

The stress in this rod, acting as a fixed-ended column, should also be found. Before this can be done the length of the rod must be assumed, and this assumption may be 2 × stroke. This would give us a rod 63 inches long for the engine under consideration. Using the formula

$$K = \frac{P}{A}\left(1 + \frac{1}{25,000}\,\frac{L^2}{\rho^2}\right),$$

where $\rho^2 = \dfrac{\text{moment of inertia of section of rod}}{\text{area of section}},$

P = total load on rod = p × area piston,

A = cross-sectional area of rod,

L = length of rod in inches,

we find the stress to be 6150 pounds per square inch. This should not exceed 7000 pounds.

The **material** used in the piston-rods should be of a good grade open-hearth steel, with or without nickel. Some manufacturers use 3½ per cent nickel steel; others use steel without nickel. In Germany it is the practice to machine the rods with a slight upward camber to compensate for the deflection due to weight of piston and rod. This is of doubtful value.

Crosshead. — All double-acting and many single-acting gas-engines are built with **crossheads.** The office of this part of the engine mechanism is to provide a flexible joint between piston-rod and connecting-rod, also to take care of connecting-rod thrust. The amount of this thrust and the area of slide necessary to take care of it may be determined as in the trunk piston. In some cases of low-speed engines, slightly higher bearing pressures may be allowed on the slides than were allowed in the trunk pistons.

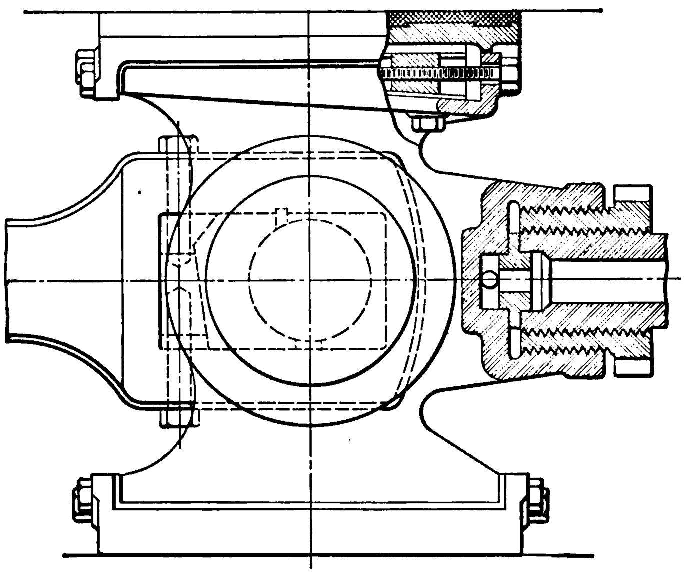


FIG. 163.

Types of **main crossheads** are shown in Figs. 163 and 164. The chief difference between the two is this: in the one shown in Fig. 163 the wear of the shoe is adjusted by means of the wedge shown; in Fig. 164 the wear is taken up by placing shims under the shoes. In both types the rod is secured in the cross-

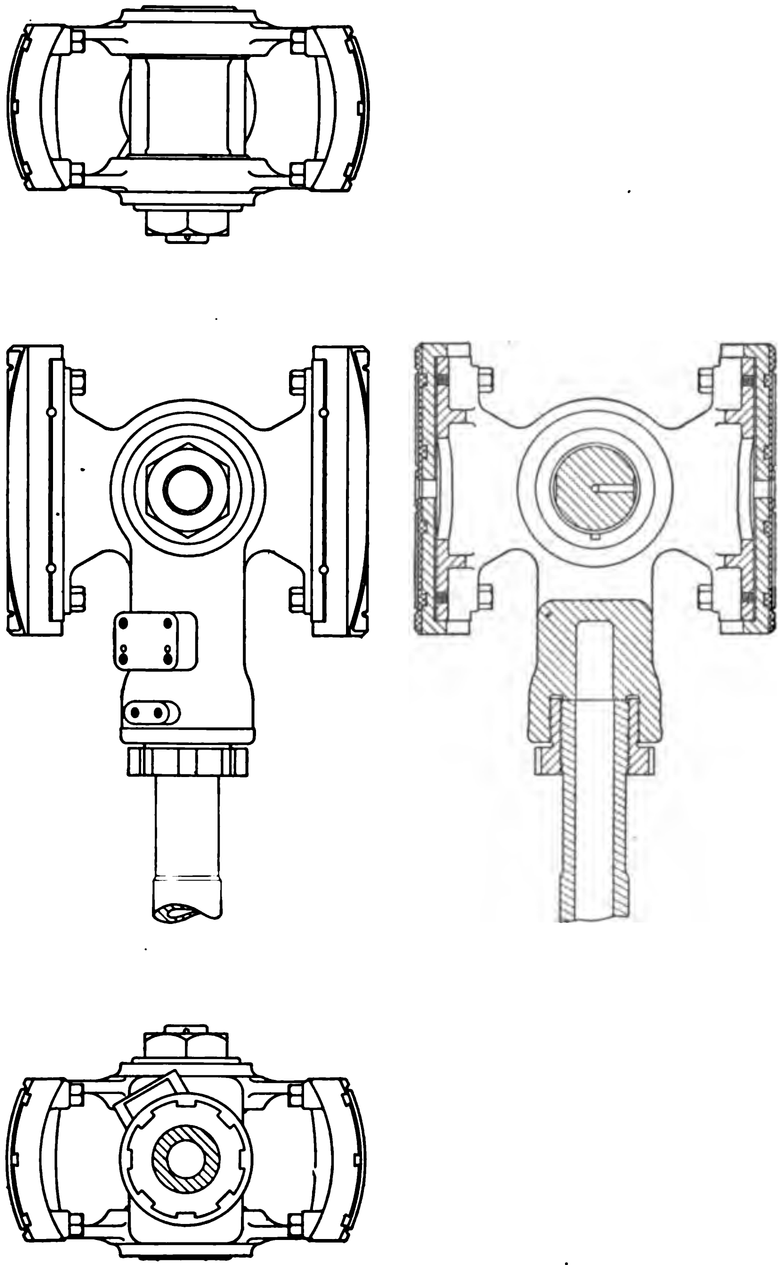
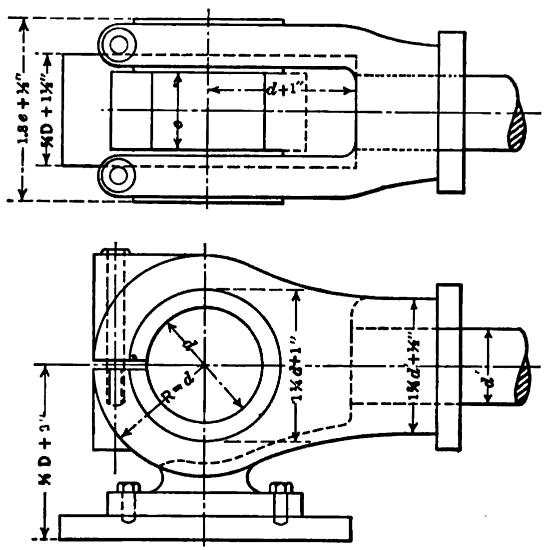


FIG. 164.

head by means of a nut threaded inside and outside. This nut is screwed into the crosshead as it is being screwed onto the rod. Since one of the threads is left hand it is obvious that the rod, crosshead and nut may all be locked together securely without employing any means to lock the nut in position.

Both the crossheads shown are of the **bored guide** type. The **marine** type, with one guide, is illustrated in Fig. 165. This type is lighter and consequently cheaper than the bored type



D = Dia. of Cylinder in inches.

Fig. 165.

but appears to lack the stiffness of the latter. The lower guide is the working surface with the engine running forward. This surface may be made ample to take any connecting-rod thrust. The upper surface takes the thrust of the rod when compression occurs in either end of the cylinder after the expansion pressure in the opposite end of the cylinder has decreased to a value below the compression pressure. The crosshead will then bear on the upper guide or slide. This slide is necessarily smaller than the bottom slide and the wear is often excessive. Premature ignition also puts a greater load on the upper slide than it should carry. For these reasons many manufacturers prefer the bored type of crosshead for double-acting engines.

The **stresses** in a crosshead are very complex and it is impossible to get a satisfactory solution, so that rational design gives way to empirical formula and data based on experience. The proportions given in Fig. 165 are applicable to many cases.

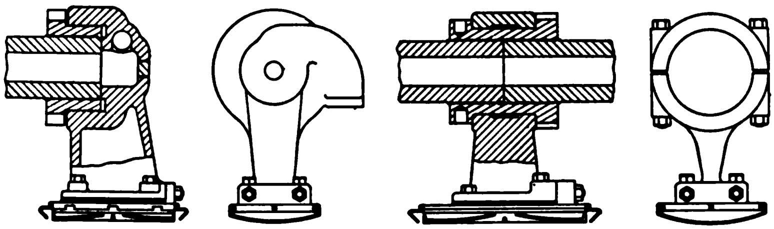


FIG. 166.

Crossheads of either type should be made of steel castings if possible, as they may be made lighter and more reliable than if made of cast-iron.

The **intermediate crosshead** of a tandem engine is illustrated

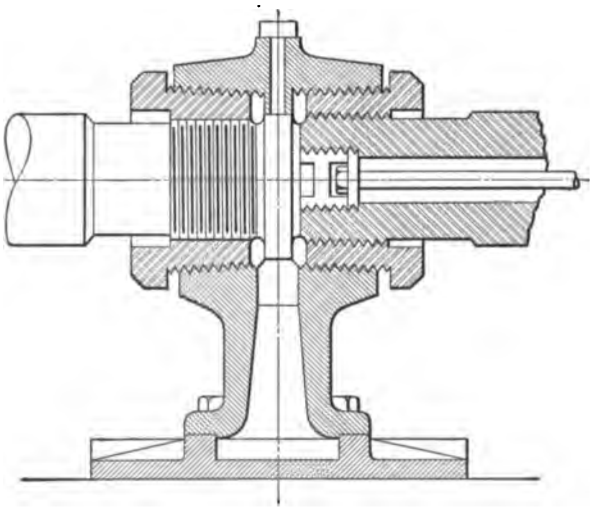


FIG. 167.

in Figs. 166 and 167. This crosshead is nothing more than a coupling for the piston-rods of the two cylinders with slide area enough to carry the weight of one piston and the rod reaching from piston to piston. The pressure per square inch on the slide

will be less than on the main crosshead. In the crosshead in Fig. 166 a single nut is used to pull the ends of the rod together. This nut is right and left hand. After the rods are thus joined, the split crosshead boss is attached by clamping. The crosshead illustrated in Fig. 167 has a solid boss and the ends of the rods are forced into this boss by a separate nut on each rod. The outside and inside threads of these nuts are right and left hand respectively, or vice versa.

This intermediate crosshead is used sometimes for admitting cooling water to the rods, in which case the water flows from this crosshead in both directions and is discharged at the main and tail crossheads. In other cases the water enters the main crosshead and passes through the pistons and rods in series, being discharged at the tail crosshead. The intermediate and **tail crossheads** in Fig. 166 are from an engine cooled in this way. The discharge passage on the tail crosshead is plainly shown. In the tail crosshead in Fig. 168, the water is discharged from a hole in the rod proper, shown ahead of the crosshead. A clamp is placed around the rod at this hole, the clamp having a passage to conduct the water to the waste passage.

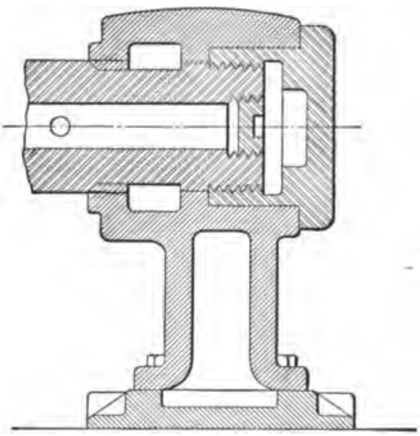


FIG. 168.

Both the intermediate and tail crossheads are simple castings of iron or steel. If of steel they may be made lighter than iron, but, as is true of most steel castings, they should be annealed. These parts are designed empirically and are made heavier than is necessary, usually. The thickness of metal should be ample for stiffness but care should be taken to keep the weight as small as possible. Adjustment is usually secured by means of shims.

The Connecting-rod. — Connecting-rods are made of steel forgings or castings. The wrist-pin and crank-pin ends should both be made adjustable so as to permit taking up the wear in

those bearings. In small engines the wrist- or piston-pin end is often made in the form of a solid eye with a bronze bushing. This is done because it is sometimes inconvenient to allow for adjustment on account of lack of space for bolts, keys, cotters, etc.

The design of the connecting-rod involves complicated calculations if exact results are to be expected. The chief complication arises when the stresses due to inertia of the rod are to be ascertained. The rod should be designed first, for direct compression at the small or piston end; second, as a pin-ended column in plane of rotation; third, for stress due to whipping action or bending due to inertia of rod.

It is quite common practice to make the connecting-rod rectangular in section, the larger dimension being that in the plane of rotation. In this case the rod should be investigated as a fixed-ended column in a plane normal to the plane of rotation.

The first step in getting the actual dimensions of the body of the rod is to determine the size at the small end for direct compression. The maximum axial force along the rod in a gas-engine is with the piston on the dead center. This force will be the area piston multiplied by the explosion pressure, when the engine first starts. After the engine gets up to speed this force will be diminished by the force of the inertia of the reciprocating parts. This force of inertia should not be taken into account, however.

The area of the small end will be the total force P divided by the allowable fibre stress for steel in alternate tension and compression. About 6000 pounds is a good value to use. If the rod is to be round, the following equation may be used to give the diameter:

$$\frac{P}{K} = \frac{\pi d^2}{4}, \tag{1}$$

$$d = \sqrt{\frac{4\,P}{\pi K}}. \tag{2}$$

If the rod is to be rectangular in section, the depth should be from $1\frac{1}{2}$ to 1.8 times the width. If we assume b, the depth, to be 1.75 times a, the width of the section, then

$$\frac{P}{K} = ab = 1.75\,a^2, \tag{3}$$

$$a = \sqrt{\frac{P}{1.75\,K}}. \tag{4}$$

In designing the rod as a **column** in the plane of rotation, we may use the Gordon formula

$$K = \frac{P}{A}\left(1 + C\frac{L^2}{\rho^2}\right), \tag{5}$$

where P is the load on the column, A the sectional area, L the length in inches, ρ the radius of gyration, and C is equal to $\frac{4}{25{,}000}$ for a pin-ended column. If the width of the section a be kept constant throughout the length of the rod, the other dimension b' will be the only unknown and will occur in the first power in A and in the second power in ρ^2. This dimension b' will be the depth of the rod halfway between the wrist-pin and crank-pin. The rough dimensions of the rod will be as shown in Fig. 169. L,

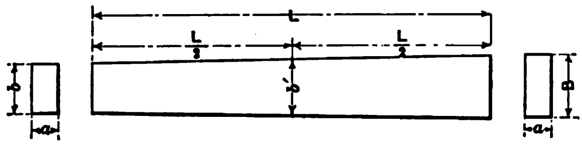


FIG. 169. — Connecting-rod Body.

the length, will be 5, 5½ or 6 times the crank radius, depending on the ratio of $\frac{L}{R}$ chosen for the engine. The depth at the center of the crank-pin will be

$$B = b + 2\,(b' - b) = 2\,b' - b. \tag{6}$$

After the stub ends have been designed the dimensions b and B ends will be modified to suit the end construction, but the dimensions of the rod body will not change. The stress in a rod of uniform cross-section due to whipping, or, in other words, due to the inertia of the rod in the plane of rotation, may be found by the formula derived by Bach,

$$K_b = 0.000002\,N^2 r A w \frac{L^2}{Z}, \tag{7}$$

where

N = revolutions per minute,
r = radius of crank in inches,
A = area mean section of rod in square inches,
w = weight of rod material in pounds per cubic inch,
L = length of rod in inches,
Z = section modulus of rod.

For a rectangular rod $Z = \frac{bh^2}{6}$, then for a rod of that shape

$$K_b = 0.000012\, N^2 rw \frac{L^2}{b'}, \qquad (8)$$

where b' is the depth of rod at the middle section.

In order to get the total stress in the rod the stress as found above should be added to that for direct tension or compression.

$$K_{\text{total}} = K_c + K_b = \frac{P}{ab'} + K_b. \qquad (9)$$

The stress in the rod as a column at right angles to the plane of rotation will be

$$K' = \frac{P}{A}\left(1 + \frac{1}{25{,}000}\,\frac{L^2}{\rho'^2}\right), \qquad (10)$$

where $$\rho'^2 = \frac{\frac{a^3b'}{12}}{ab'} = \frac{a^2}{12}.$$

Example. — To illustrate the design of the body of the rod we will assume a cylinder diameter of 21 inches, a stroke of 30 inches, an explosion pressure of 425 pounds per square inch and 150 r.p.m. The total force P will be 147,000. If we assume

$$b = 1.7\,a, \quad \text{then} \quad \frac{147{,}000}{6000} = 1.7\,a^2,$$

$$a = \sqrt{\frac{147{,}000}{6000 \times 1.7}} = 3.82 = 3\tfrac{13}{16} \text{ inches},$$

$$b = 1.7 \times 3.82 = 6\tfrac{1}{2} \text{ inches}.$$

To find b', by (5)

$$6000 = \frac{147{,}000}{3.82 \times b'}\left(1 + \frac{4}{25{,}000} \times \frac{90 \times 90 \times 12}{b'^2}\right),$$

$$0.1555\, b' = 1 + \frac{15.43}{b'^2},$$

$$0.1555\, b'^3 - b'^2 = 15.43.$$

Solving this equation for b' we find the depth of the rod at the middle to be 8 inches. By proportion we find the depth at the crank-pin to be 9 inches.

The stress due to whipping will be approximately

$$K_b = 0.000012 \times 150 \times 150 \times 15 \times 0.282 \times \frac{90 \times 90}{8}$$
$$= 1160 \text{ pounds per square inch.}$$
$$K_{\text{total}} = \frac{147{,}000}{8 \times 3.82} + 1160 = 5960 \text{ pounds per square inch.}$$

The maximum stress due to whipping does not actually occur at the center of the rod but at a place nearer the crank-pin, where the rod is of larger section. But as this location is difficult to determine before the actual shape and weight of the rod is known, we assume it to be at the middle of the rod, which is on the side of safety.

The stress in the rod assuming the rod to be a fixed-ended column in a plane normal to the plane of rotation will be

$$K' = \frac{147{,}000}{3.82 \times 8}\left(1 + \frac{1}{25{,}000} \times \frac{90 \times 90 \times 12}{3.82 \times 3.82}\right)$$
$$= 6080 \text{ pounds per square inch.}$$

Stub Ends. — A connecting-rod for a large double-acting gas-engine is illustrated in Fig. 170. The rod and straps are forged

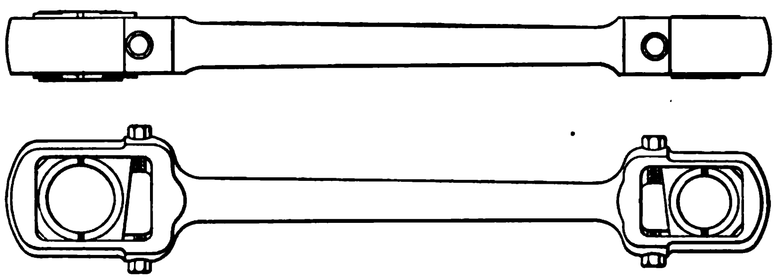


FIG. 170. — Forged-steel Connecting Rod.

in one piece. Babbitted shells or boxes are used at both ends, adjustment being secured by means of wedges. In this case adjustment at either end tends to lengthen the rod. The rod shown in Fig. 171 is a steel casting, with straps cast solid with the rod. The babbitted shells are wedge-adjusted in this case also, but adjustment tends to lengthen the rod at the wrist-pin end and shorten it at the crank-pin end. One noticeable feature of this rod is the I section which is difficult to secure in a forged rod but not difficult in a cast rod. Automobile rods which are I shaped are usually drop-forged.

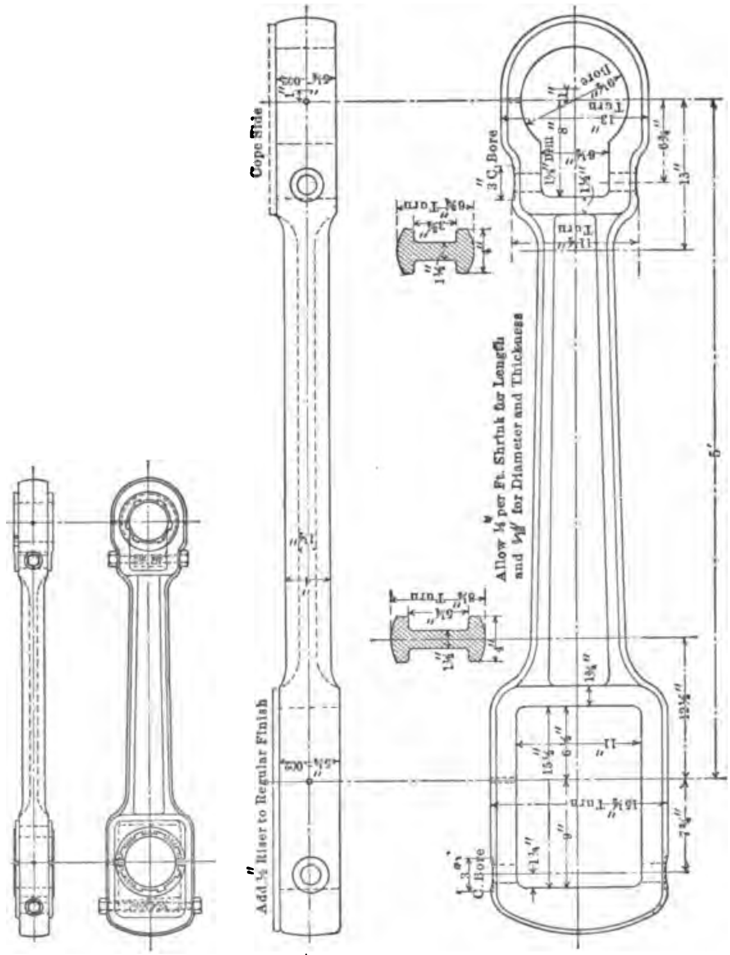


FIG. 171. — Cast-steel Connecting-rod.

The diameter and length of the wrist-pin have already been computed. These two dimensions form a basis for the calculations that follow. The width of the bearing is obviously equal to the effective length of the pin. The width of the straps that enclose the bearing may be from ½ to 1½ inches less than the effective length of the pin. Thus in Fig. 172, c may be made from $e - \frac{1}{2}$ inch to $e - 1\frac{1}{2}$ inches. The thickness of the strap a may be found by computing each side to take half the total piston pres-

sure P. Thus $\frac{P}{2K} = a\,(c - \delta)$. In this case a stress of 7000 pounds might be used for K.

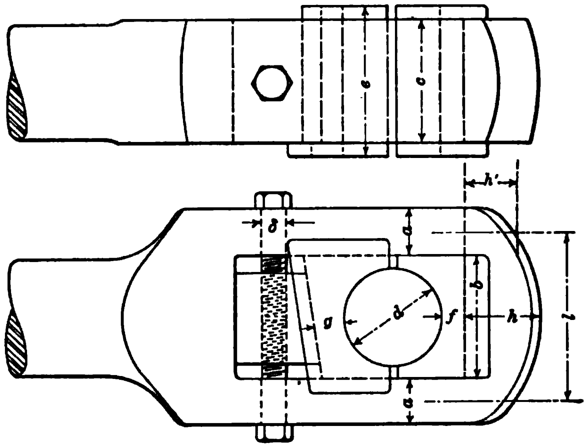


FIG. 172. — Wrist-pin Stub End.

To find the thickness h at the end of the rod we must first find b. The value b may be assumed to be $d + (\frac{1}{2}$ to $1\frac{1}{2}$ inches), the constant depending on the size of the engine. The end of the rod is similar to a beam with a uniform load whose total value is P. The bending moment is then $\frac{Pl}{8}$, where $l = b + a$. The value h will be found from the equation

$$\frac{Pl}{8} = \frac{Kch^2}{6}.$$

The stress may run up to 10,000 pounds in this part of the rod, if the above equation be used. The values of f and g vary between rather wide limits. Where bronze boxes are used these dimensions may be less than where babbitted shells are used. We might set the limits at $\frac{1}{2}$ inch and $1\frac{1}{2}$ inches, without being far wrong. The taper of the wedge should be one in six, and the diameter of adjusting screws will fall between $\frac{3}{4}$ inch and $1\frac{1}{2}$ inches.

The design of the crank-pin closed-strap end is not different from that given above. If a marine-end be employed, the design changes slightly. In Fig. 173 is shown a detail of the marine end such as is usually employed when a shaft with a center-crank

is used. Before the design can be carried to completion it will be necessary to find the approximate size of the crank-pin. Fig. 174

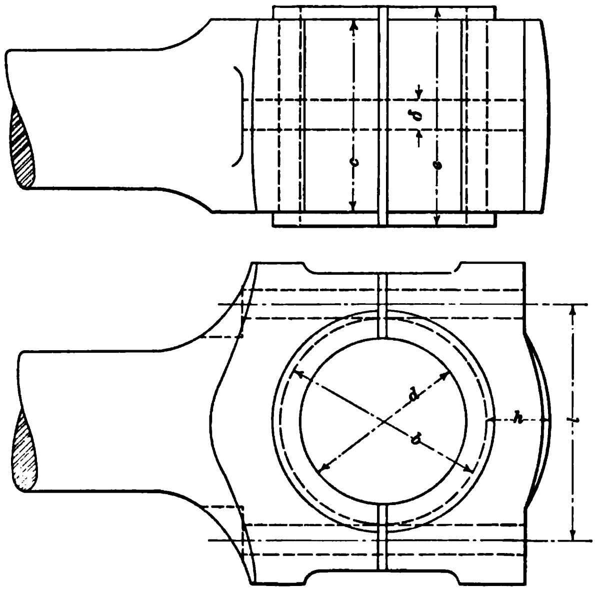


FIG. 173. — Crank-pin Stub End — Marine Type.

illustrates this step. The distance between the centers of the bearings may be assumed to be from $1.8\,D$ to $2.2\,D$, where D is

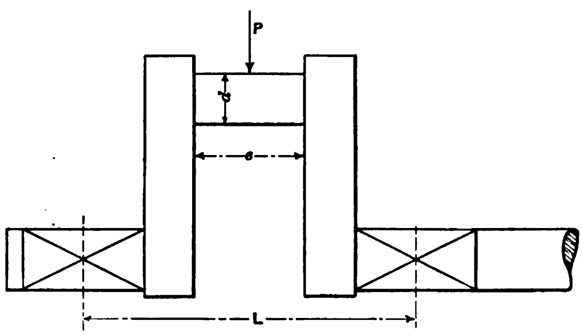


FIG. 174.

the cylinder diameter in inches. The diameter of the pin may be found from

$$\frac{PL}{4} = \frac{K\pi d^3}{32},$$

whence $$d = \sqrt[3]{\frac{32\,PL}{4\,K\pi}},$$

where $K = 12{,}000$ pounds per square inch. The length e may be made the same as the diameter. The bearing pressure $\frac{P}{d \times e}$ should be found and should fall between the values 1200 and 1500 pounds per square inch.

The design of a side crank-pin is a little different from that of a center crank-pin. It is convenient to assume the ratio of $\frac{e}{d}$, Fig. 175, in starting the design of a side crank-pin. Then the bending moment on the pin may be taken as $\frac{3}{4}\,Pe = \frac{3}{4}\,Pkd$, where $\frac{e}{d} = k$. Theoretically the pressure is evenly distributed over the length of the pin, giving a bending moment of $\frac{1}{2}\,Pe$. Actually the pressure may not be evenly distributed. Also, it would be too severe to say that the pressure is exerted at the end of the pin. Hence we use the mean value $\frac{3}{4}\,Pe$.

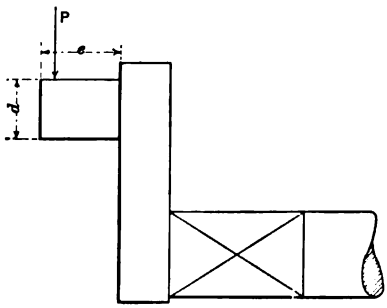


Fig. 175.

Then $$\frac{3}{4}\,Pkd = \frac{K\pi d^3}{32},$$

$$d = \sqrt{\frac{3\,P \times k \times 32}{4\,\pi K}}.$$

Here again the bearing pressure should be investigated and should fall between 1200 and 1500.

After the size of the pin has been determined, the other dimensions shown in Fig. 173 may be found. Thus, $b = d +$ ($\frac{1}{2}$ to $1\frac{1}{2}$ inches), and $l = b + \delta$. To get δ, assume each bolt to take its share of the total pressure on the piston.

$$\frac{P}{2} = \frac{K\pi d'^2}{4}, \qquad d' = \sqrt{\frac{4\,P}{2\,\pi K}},$$

where d' = diameter of bolt at root of thread,
$K = 12{,}000$.

The diameter of bolt δ will be d' + twice the depth of the threads. Fine threads are usually used on these bolts and the body of the bolt is turned down to the diameter of the root of the thread as shown in Fig. 176. This is done to prevent local strains. The part of the bolt that bears in the rod and cap should be a tight fit to line the parts up properly. These bolts are usually ground to a fit.

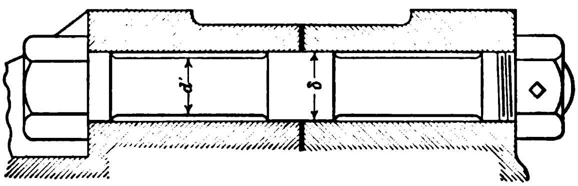


FIG. 176.

The nuts should be locked securely. The one in Fig. 176 is locked with a small set-screw tapped in the side. On smaller bolts double nuts may be used. Lock-plates are sometimes employed on large nuts.

It is best to get the bolts as close together as possible and in order to do this the bearing shell is cut away. In some cases the bolts are brought in until they just clear the pin. The thickness of the shell including babbitt should be from $\frac{3}{4}$ to $1\frac{1}{2}$ inches, depending on the size, when the shell is of cast steel. The thickness of the cap h may be computed from

$$\frac{P}{2}\left(\frac{l}{2} - \frac{b}{4}\right) = \frac{Kch^2}{6},$$

where c is from $\frac{1}{2}$ to $1\frac{1}{2}$ inches less than e.

The calculations above apply only to double-acting engines. The force that acts on the straps or crank-pin bolts of the connecting-rod for a single-acting engine is the inertia of the reciprocating parts. This force is so much less than the pressure due to the explosion that parts designed with it as a load would be very light compared to parts designed for the explosion pressure.

Fig. 177 illustrates a connecting-rod for a single-acting Diesel engine. The crank-pin cap is noticeably light, as are the bolts, compared to a double-acting engine. Another peculiarity is the wedge adjustment. On account of lack of room in the piston, the wedge is put in at right angles to the plane of the rod. This gives more room for using a wrench but shortens up the wedge.

Still another type of adjustment is shown in Fig. 178 which is the rod used in the Foos vertical multi-cylinder engine. This arrangement brings the adjustment outside the piston but costs

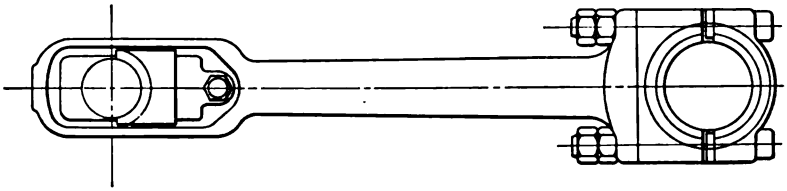


FIG. 177. — Connecting-rod for Single-acting Diesel Engine.

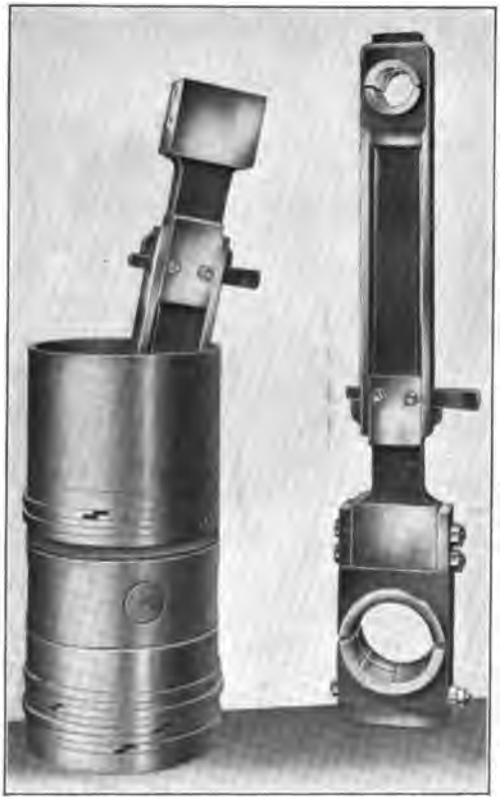


FIG. 178. — Foos Connecting-rod and Piston.

more to make than the one in Fig. 177. The crank-pin end shows the open-strap-end adapted to the center-crank. Both these adjustments tend to shorten the rod.

Some engineers do not use the method given on pages 308 and 311 for finding the thickness h of the connecting-rod end. Instead they design the end for shear, thus:

$$h' = \frac{P}{2K_s c},$$

where h' is the minimum thickness of the end of the rod, Fig. 172, P is the total maximum piston pressure in pounds, c is the width of the rod in inches and K_s is the allowable shearing stress, about 4000 pounds per square inch.

The thickness h at the middle is made from $\frac{1}{2}$ to 1 inch greater than h', depending on the size of the rod.

EXERCISES

1. Find the thickness of the head of a trunk piston 20 ins. diameter, pressure 400 lbs. per sq. in. Material cast-iron.

2. Find the stress in the head of a trunk piston 16 ins. diameter, $1\frac{3}{8}$ ins. thick, 400 lbs. press.

3. Find the thickness of cast-iron piston rings for a 20-in. diameter cylinder, pressure between ring and cylinder 4 lbs. per sq. in. Find stress due to slipping ring into groove.

4. Find thickness of cast-iron piston rings for 20-in. diameter cylinder, pressure between cylinder and wall 5 lbs. per sq. in. Find stress due to slipping ring into groove.

5. Find the amount cut out of the rings in (3) and (4).

6. Design the piston-pin for the piston in (1), using bearing pressure and stress. Assume effective length of pin to be one-half cylinder diameter.

7. Design the piston-pin for a Diesel engine which has a cylinder 16 ins. diameter. Pin to be 8 ins. long.

8. Design the piston-rod for cylinder 20 ins. diameter, assuming length to be $2\frac{1}{2} \times$ diameter of cylinder. Section of rod to be solid circle. Maximum piston pressure 380 lbs. per sq. in.

9. Design piston-rod for a cylinder 16 ins. diameter, length of rod to be $3 \times$ diameter of cylinder. Rod to have hole whose diameter is one-half outside diameter of rod. Maximum piston pressure 400 lbs. per sq. in.

10. Find size of crosshead shoe for engine with cylinder 20 ins. diameter, maximum piston pressure 375 lbs. per sq. in. Vertical component of connecting-rod thrust to be $\frac{1}{10}$ total pressure on piston. Length of shoe $2 \times$ width.

11. Design rectangular connecting-rod for cylinder in Prob. 9, length of rod to be $4 \times$ diameter cylinder. Depth of rectangular section at the small end to be $1\frac{3}{4} \times$ the constant width. 175 r.p.m.

12. Find stress in rod in Prob. 11 in plane at right angles to plane of oscillation. Find stress due to whipping, if r.p.m. = 175, stroke = 24 ins.

13. Find the diameter of connecting-rod bolts, crank-pin end, of double-acting engine with 18-in. diameter cylinder, 375 lbs. maximum pressure.

14. Design crank-pin of center-crank type for engine in Prob. 13. Distance between center lines of bearings to be $2.1 \times$ diameter cylinder.

15. Design crank-pin of side-crank type for engine in Prob. 13.

CHAPTER XXI

INERTIA OF RECIPROCATING PARTS — NET EFFORT DIAGRAMS — ROTATIVE EFFORT DIAGRAMS — WEIGHT OF FLYWHEEL — CONSTRUCTION OF FLYWHEEL — VELOCITY AND DISPLACEMENT DIAGRAMS

Weight of Reciprocating Parts. — In using the weight of reciprocating parts to find the effect of their inertia on the crank-pin pressure, the weight of the complete connecting-rod is usually included as a reciprocating part, though theoretically about half the weight of the rod has a rotating and not a reciprocating motion. In a tandem double-acting engine the weight of reciprocating parts will include the pistons, piston-rods, connecting-rod, main crosshead, intermediate crosshead and coupling, tail crosshead, water connections and cooling water in pistons and rods. The weights of the various parts may be computed from sketches made for each part as it is designed. For this computation the weight of cast-iron may be taken as 0.26 and steel 0.282 pound per cubic inch. Bronze weighs about 0.32 and brass 0.30 pound per cubic inch.

For approximate computations it is sometimes an advantage to be able to assume a weight of reciprocating parts. These may be assumed for the various engine types as shown in Table XIII.

Inertia Due to Reciprocating Parts. — The reciprocating parts of an engine have a velocity of zero at the ends of the stroke. In order to accelerate these parts up to their maximum velocity, which occurs somewhere near mid-stroke, the force of acceleration must be applied. The force of acceleration is supplied by the gas pressure, and reduces the effective pressure of the gas at the beginning of the stroke and increases it at the end of the stroke. In order to find an expression for the force of acceleration it is necessary to find first the expression for velocity. In Fig. 179 is shown the graphical construction which gives v, the piston velocity for the crank angle θ, when V is the constant velocity of the crank-pin.

TABLE XIII

WEIGHT OF RECIPROCATING PARTS PER SQUARE INCH OF PISTON

Kind of engine.	Weight per square inch of piston.
Single-acting:	
Trunk piston $L < 1.75\ D$	3.8 –4.2
Trunk piston $L > 1.75\ D$	4.2 –5.0
With crosshead	5.8 –7.2
Small high-speed automobile engines	0.45–0.75
Double-acting four-cycle:	
Single-cylinder	11–18
Tandem	18–25
Double-acting two-cycle	16–21

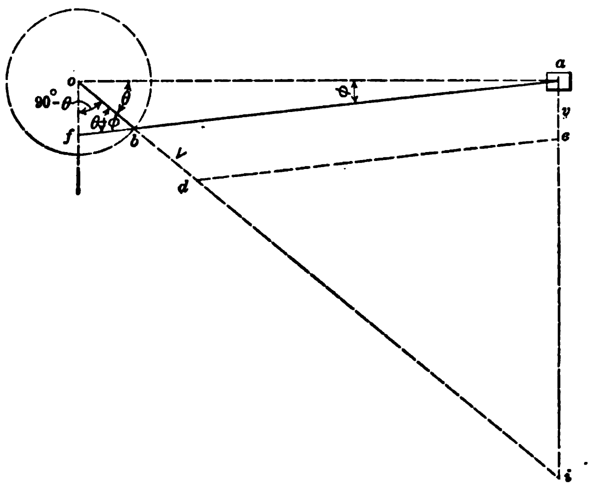


FIG. 179.

By similar triangles it may be shown that if the crank radius *ob* is assumed as the constant velocity of the crank-pin, the intercept on the vertical through *o* of the connecting-rod or the connecting-rod produced will give the velocity of the piston at that instant. Thus, in Fig. 179, *of* is the velocity of the reciprocating parts for the crank angle θ, when *ob* is the velocity of the crank-pin.

$$\frac{of}{ob} = \frac{\sin obf}{\sin ofb} = \frac{\sin(\theta + \phi)}{\cos\phi}.$$

For the outstroke

$$\frac{v}{V} = \frac{\sin(\theta + \phi)}{\cos\phi}.$$

For the return stroke

$$\frac{v'}{V} = \frac{\sin(\theta' - \phi')}{\cos\phi'}.$$

Then

$$\left.\begin{aligned} v &= V(\sin\theta + \cos\theta \ \tan\phi), \\ v' &= V(\sin\theta' - \cos\theta' \tan\phi'). \end{aligned}\right\} \tag{1}$$

When ϕ is a small angle, as is always the case in practical problems,

$$\tan\phi = \sin\phi = \frac{\sin\theta}{n}, \text{ approximately,}$$

where $n = \frac{L}{r}$ = length of connecting-rod divided by length of crank. Hence

$$\left.\begin{aligned} v &= V\left(\sin\theta + \frac{\sin 2\theta}{2n}\right), \\ v' &= V\left(\sin\theta' - \frac{\sin 2\theta'}{2n}\right). \end{aligned}\right\} \tag{2}$$

If α = acceleration in feet per second per second, then

$$\alpha = \frac{dv}{dt} = \frac{dv}{d\theta}\frac{d\theta}{dt} = \omega\frac{dv}{d\theta} = \frac{V}{r}\frac{dv}{d\theta}.$$

Then, using equation (2),

$$\left.\begin{aligned} \alpha &= \frac{V^2}{r}\frac{d}{d\theta}\left(\sin\theta + \frac{\sin 2\theta}{2n}\right) = \frac{V^2}{r}\left(\cos\theta + \frac{\cos 2\theta}{n}\right), \\ &\frac{V^2}{r}\frac{d}{d\theta'}\left(\sin\theta' - \frac{\sin 2\theta'}{2n}\right) = \frac{V^2}{r}\left(\cos\theta' - \frac{\cos 2\theta'}{n}\right). \end{aligned}\right\} \tag{3}$$

The force of acceleration is the product of the mass into the acceleration, or $F = M\alpha$. For the sake of convenience we use the weight of reciprocating parts per square inch of piston area. Then

$$\left.\begin{aligned} F &= \frac{W}{Ag}\frac{V^2}{r}\left(\cos\theta + \frac{\cos 2\theta}{n}\right), \\ F' &= \frac{W}{Ag}\frac{V^2}{r}\left(\cos\theta' - \frac{\cos 2\theta'}{n}\right), \end{aligned}\right\} \tag{4}$$

where W = total weight of reciprocating parts in pounds,
A = piston area in square inches,
V = crank-pin velocity in feet per second,
r = crank radius in feet,
g = 32.2.

TABLE XIV

Values of Terms in Expression for Force of Inertia

θ degs.	$\cos\theta$	$\cos 2\theta$	$\frac{L}{r}=4$		$\frac{L}{r}=4\frac{1}{2}$		$\frac{L}{r}=5$		$\frac{L}{r}=5\frac{1}{2}$		$\frac{L}{r}=6$	
			$\frac{\cos 2\theta}{n}$	$\cos\theta+\frac{\cos 2\theta}{n}$	$\frac{\cos 2\theta}{n}$	$\cos\theta+\frac{\cos 2\theta}{n}$	$\frac{\cos 2\theta}{n}$	$\cos\theta+\frac{\cos 2\theta}{n}$	$\frac{\cos 2\theta}{n}$	$\cos\theta+\frac{\cos 2\theta}{n}$	$\frac{\cos 2\theta}{n}$	$\cos\theta+\frac{\cos 2\theta}{n}$
0	1.0000	1.0000	0.2500	1.2500	0.2222	1.2222	0.2000	1.2000	0.1818	1.1818	0.1667	1.1667
15	0.9659	0.8660	0.2165	1.1824	0.1924	1.1583	0.1732	1.1381	0.1575	1.1270	0.1443	1.1102
30	0.8660	0.5000	0.1250	0.9910	0.1111	0.9771	0.1000	0.9660	0.0909	0.9569	0.0833	0.9493
45	0.7071	0.0000	0.0000	0.7071	0.0000	0.7071	0.0000	0.7071	0.0000	0.7071	0.0000	0.7071
60	0.5000	−0.5000	−0.1250	0.3750	−0.1111	0.3889	−0.1000	0.4000	−0.0909	0.4191	−0.0833	0.4167
75	0.2588	−0.8660	−0.2165	0.0423	−0.1924	0.0664	−0.1732	0.0856	−0.1575	0.1013	−0.1443	0.1145
90	0.0000	−1.0000	−0.2500	−0.2500	−0.2222	−0.2222	−0.2000	−0.2000	−0.1818	−0.1818	−0.1667	−0.1667
105	−0.2588	−0.8660	−0.2165	−0.4753	−0.1924	−0.4512	−0.1732	−0.4320	−0.1575	−0.4163	−0.1443	−0.4031
120	−0.5000	−0.5000	−0.1250	−0.6250	−0.1111	−0.6111	−0.1000	−0.6000	−0.0909	−0.5909	−0.0833	−0.5833
135	−0.7071	0.0000	0.0000	−0.7071	0.0000	−0.7071	0.0000	−0.7071	0.0000	−0.7071	0.0000	−0.7071
150	−0.8660	0.5000	0.1250	−0.7410	0.1111	−0.7549	0.1000	−0.7660	0.0909	−0.7751	0.0833	−0.7827
165	−0.9659	0.8660	0.2165	−0.7494	0.1924	−0.7735	0.1732	−0.7927	0.1575	−0.8084	0.1443	−0.8216
180	−1.0000	1.0000	0.2500	−0.7500	0.2222	−0.7778	0.2000	−0.8000	0.1818	−0.8182	0.1667	−0.8333

In order to get the values of F for different crank angles it is well to make a table having columns for θ, $\cos\theta$, $\cos 2\theta$, $\frac{\cos 2\theta}{n}$, $\cos\theta + \frac{\cos 2\theta}{n}$ and $\frac{WV^2}{Agr}\left(\cos\theta + \frac{\cos 2\theta}{n}\right)$.

The first five columns will have constant values for a given value of n, and values for these are given in Table XIV for five different values of n. The last column of the table, that is,

$$\frac{WV^2}{Agr}\left(\cos\theta + \frac{\cos 2\theta}{n}\right),$$

will give the values in pounds per square inch for the ordinates of the inertia diagram, the use of which is explained later.

Net Effort Diagrams. — The effect of the inertia of the reciprocating parts is to decrease the pressure at the beginning of the stroke and increase the pressure at the end of the stroke when the pressure in the cylinder has decreased due to expansion of the gas, thus tending to make the turning effort on the crank-pin more uniform.

The actual use of the inertia diagram is illustrated in Figs. 180 and 181 which are diagrams showing the combination of piston pressure with the inertia. In order to get these diagrams the lines on the indicator card are plotted in sequence, beginning with the expansion or working stroke, the total length of the diagram representing 720 degrees of the crank travel. The values of the ordinates of the inertia diagram which appear in the table as positive are plotted below the base line since they are retarding pressures, and those appearing as negative are plotted above the base line.

The pressure lines in Fig. 180 are from a single-acting, four-cycle, single-cylinder engine. The only positive piston pressure, that is, pressure which results in a positive turning moment on the crank-pin, occurs during the expansion stroke. This line will give a large area on the rotative effort diagram, the other 540 degrees of this diagram being influenced chiefly by the inertia diagram, with compression on the last stroke.

The pressure lines in Fig. 181 are from a double-acting, single-cylinder, four-cycle engine. Since the explosion strokes are but 180 degrees apart, we must expect two large areas on the rotative effort diagram close together. The effect of this will be discussed in a later paragraph. The pressure lines in Figs. 180 and

181 are taken directly from the indicator cards, the ordinates corresponding to the crank angles, say every 15 degrees. Care must be taken to get the ordinates spaced in proper order. Those

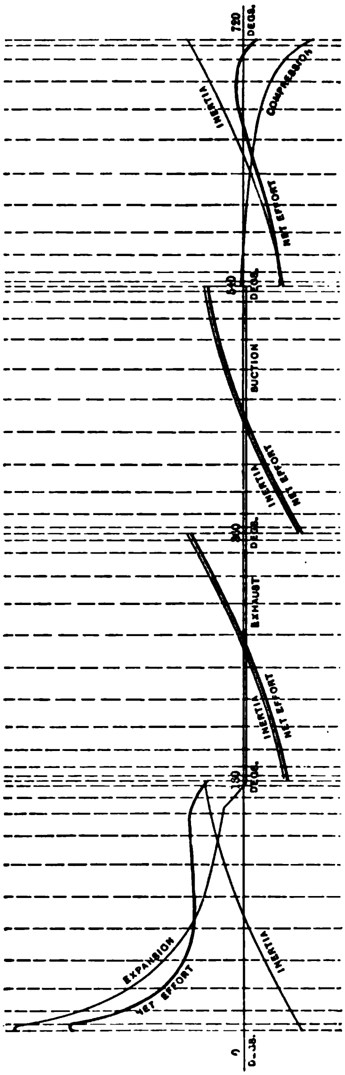


FIG. 180. — Net Effort Curve for a Four-cycle, Single-cylinder, Single-acting Gas-engine.

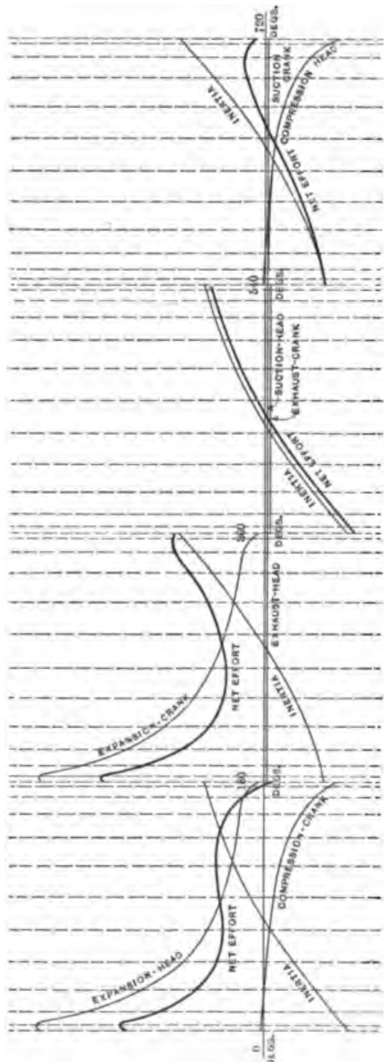


FIG. 181. — Net Effort Curve for a Four-cycle, Double-acting, Single-cylinder Gas-engine.

in the strokes from 180 to 360 degrees and from 540 to 720 degrees are in reverse order, which gives the inertia diagram the peculiar shape shown.

The **net pressure** on the piston is the difference between the pressures on the two sides of the piston occurring at the same instant. In the single-acting engine it is the difference between the pressure in the cylinder and atmospheric pressure, or it is the gauge pressure. In the double-acting engine, care must be taken to use the proper pressure as a back pressure. After the net pressure has been found, this pressure should be combined with the inertia pressure which will give the **net effort,** the pressure which is transmitted to the crank-pin. In Fig. 182 is shown a detail illustrating the manner in which these pressures should be combined. Curve 1–2 represents the expansion line for the head-end of the cylinder, 3–4 the compression line for the crank-end and 5–6 the inertia of the reciprocating parts.

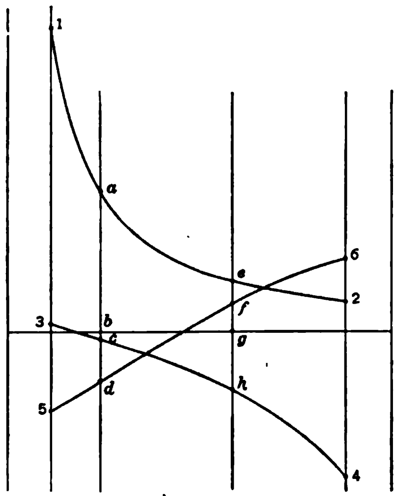


FIG. 182. — Detail of Net Effort Curve.

On the ordinate *ad* which may represent some crank angle, $ab - bc$ is the **net pressure** on the piston. The value of inertia at that point is $-bd$, so the **net effort** is $ab - bc - bd$. On the ordinate *eh*, the pressure on the expansion line is $+eg$ and that on the compression line is $-gh$, so the net pressure is $eg - gh$. The value of inertia at this point is $+fg$. Hence the net effort is $eg - gh + fg$.

If a tandem double-acting engine is under consideration there will be four pressure lines for each stroke, expansion, exhaust, suction and compression. These four may be combined as above to find the net pressure and then combined with the inertia to get the net effort.

Rotative Effort Diagram. — After the net effort on the piston has been found as in the preceding article, the rotative effort should be found. If only a few points are to be determined, a graphical construction is most convenient. Thus, in Fig. 183, if the crank is extended and *ab* is laid off equal to the net effort on piston at *c*, the intercept *cd*, found by drawing *bd* parallel to the

connecting-rod, will be the rotative effort for the crank position *oa*. *Proof:* It has been shown that when *ab* is the velocity of the crank-pin, *cd* is the velocity of the piston. Neglecting friction the work per minute will be equal at the two points. Hence the

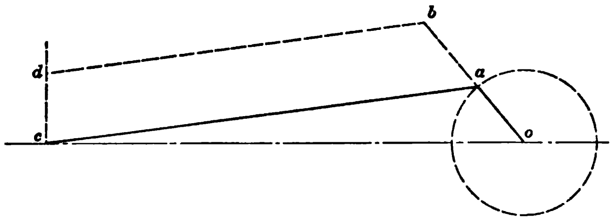


FIG. 183. — Graphical Method for Finding the Rotative Effort.

forces at the two points are inversely as the velocities, and if *cd* represents the velocity of the piston, it will also represent the working or tangential force on the crank-pin.

When a number of points are to be found, as in a rotative effort diagram, it is better to find them by calculation. In Fig. 184 we

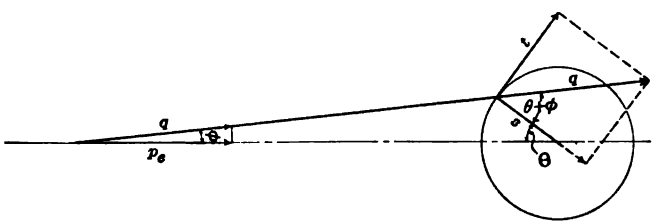


FIG. 184.

have q, the thrust along the rod, equal to $p_e \sec \phi$, where p_e is the net effort on the piston taken from the net effort diagram for that particular crank angle. Resolving q along and at right angles to the crank, we find t, the rotative or tangential effort, and s, the radial thrust.

$$q = p_e \sec \phi,$$

$$t = q \cos [90 - (\theta + \phi)] = q \sin (\theta + \phi)$$

$$= p_e \sec \phi \sin (\theta + \phi) = p_e \frac{\sin (\theta + \phi)}{\cos \phi}, \qquad (1)$$

$$\sin (\theta + \phi) = \sin \theta \cos \phi + \cos \theta \sin \phi.$$

Then $$\frac{\sin (\theta + \phi)}{\cos \phi} = \sin \theta + \frac{\cos \theta \sin \phi}{\cos \phi}. \qquad (2)$$

But
$$\sin\phi = \frac{\sin\theta}{n},$$
$$\cos\phi = \sqrt{1 - \frac{\sin^2\theta}{n^2}}.$$

Hence
$$\frac{\sin(\theta+\phi)}{\cos\phi} = \sin\theta\left(1 + \frac{\cos\theta}{\sqrt{n^2 - \sin^2\theta}}\right).$$

Or
$$t = p_e \sin\theta\left(1 + \frac{\cos\theta}{\sqrt{n^2 - \sin^2\theta}}\right). \qquad (3)$$

Where t = tangential or rotative effect and p_e = pressure from net effort diagram.

In order to use equation (3) it is convenient to have a table containing the values of $\sin\theta\left(1 + \frac{\cos\theta}{\sqrt{n^2 - \sin^2\theta}}\right)$ for different values of θ and n. The values of this expression are given in Table XV for every 15 degrees and values of n of 5, 5½ and 6. The values in this table need only to be multiplied by the p_e taken from the net effort diagram for that crank angle. The values of t thus obtained should be used as ordinates for the curve as shown in Figs. 185 and 186, which are rotative effort curves for the net effort diagrams shown in Figs. 180 and 181.

TABLE XV

VALUES OF $\sin\theta\left(1 + \frac{\cos\theta}{\sqrt{n^2 - \sin^2\theta}}\right)$

Degrees.	$n=4$	$n=4\frac{1}{2}$	$n=5$	$n=5\frac{1}{2}$	$n=6$
0	0.0000	0.0000	0.0000	0.0000	0.0000
15	0.3214	0.3144	0.3088	0.3043	0.3005
30	0.6091	0.5968	0.5870	0.5790	0.5724
45	0.8341	0.8196	0.8081	0.7988	0.7910
60	0.9769	0.9642	0.9540	0.9457	0.9390
75	1.0303	1.0227	1.0169	1.0121	1.0081
90	1.0000	1.0000	1.0000	1.0000	1.0000
105	0.9015	0.9091	0.9149	0.9197	0.9237
120	0.7551	0.7680	0.7781	0.7863	0.7930
135	0.5801	0.5946	0.6061	0.6154	0.6232
150	0.5181	0.4032	0.4130	0.4210	0.4276
165	0.1962	0.2032	0.2087	0.2133	0.2171
180	0.0000	0.0000	0.0000	0.0000	0.0000

The base of these curves is the distance moved through by the crank-pin during the time the piston moves through the distance

represented by the base of the net effort diagrams. In the case of four-cycle engines this should always be four strokes, hence the base of the rotative effort diagram will be $2\pi l$, where l is the length of the indicator card. When tandem, double-acting cylinders are being considered the rotative effort diagram need be only πl inches long. If the same pressure scale is used on the indicator and rotative effort diagrams, then the areas of both should be equal. This is a convenient check on the accuracy of the work at this point.

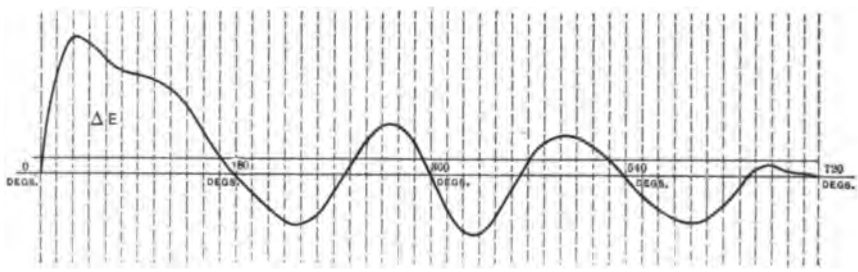


FIG. 185. — Rotative Effort Diagram for a Four-cycle, Single-cylinder, Single-acting Gas-engine.

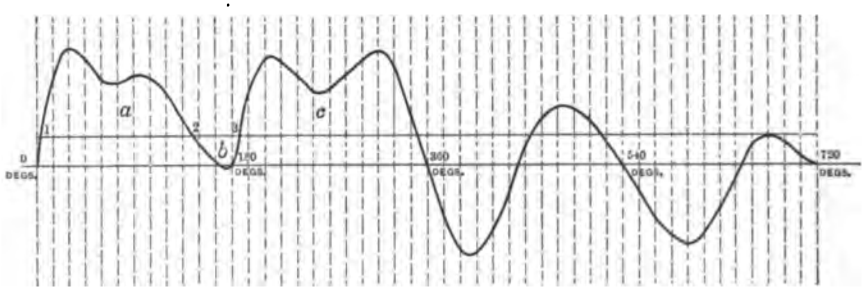


FIG. 186. — Rotative Effort Diagram for a Four-cycle, Single-cylinder, Double-acting Gas-engine.

Excess Energy. — If the area of the rotative effort diagram is found, and this area is divided by the length, the result is the mean tangential force on the crank-pin. If the actual tangential force on the crank-pin were constant there would be no need of having a flywheel. The necessity of having a flywheel is shown by the area above or below the mean tangential effort line. If we assume the power taken off the engine shaft to be constant during a cycle, then the area above or below the mean line represents the

energy absorbed and given out by the flywheel during its allowable fluctuation in speed. The maximum single area between the curve and the mean line usually is the area on which the weight of the flywheel is based. In some cases the excess area is the sum of two large areas above the mean line minus a smaller area between these two and below the line. In Fig. 185 the maximum excess area is evidently that marked ΔE. In Fig. 186 the excess area is not so evident. The flywheel increases in speed while absorbing the energy represented by a, and slows down while giving out the energy b. But as b is less than a the wheel is revolving at a higher velocity at 3 than at 1. A further increase in velocity must take place in order to absorb the energy represented by c. Hence the total energy which the flywheel must absorb during its allowable increase in speed is represented by the area $a + c - b$.

Weight of Flywheel.—The expression for kinetic energy of a body moving at a velocity of V feet per second is

$$\text{K.E.} = \tfrac{1}{2} MV^2, \tag{1}$$

where M = mass of the body. Using for M the expression $\dfrac{W}{g}$ in order to change to engineering units, the change in kinetic energy for a change in velocity from V_1 to V_2 will be

$$\begin{aligned} \Delta E &= \frac{W}{2g}(V_1^2 - V_2^2) \\ &= \frac{W\,4\pi^2\rho^2}{2\,g\,3600}(N_1^2 - N_2^2) \\ &= \frac{W\,4\pi^2\rho^2}{2\,g\,3600}(N_1 - N_2)(N_1 + N_2), \end{aligned} \tag{2}$$

where N_1 is the maximum speed and N_2 the minimum speed in revolutions per minute and ρ is the radius of gyration in feet.

$$N_1 + N_2 = 2N,$$

$$\frac{N_1 - N_2}{N} = \frac{1}{k}, \text{ the coefficient of fluctuation.}$$

Then
$$\Delta E = \frac{1}{2}\,\frac{W}{g}\,\frac{4\pi^2\rho^2 \times 1 \times 2N^2}{3600 \times k}, \tag{3}$$

when N = mean speed in revolutions per minute.

Hence
$$W = \frac{\Delta E \times g \times 900 \times k}{\pi^2\rho^2 N^2}. \tag{4}$$

The weight W thus obtained is the weight of the wheel in pounds at the radius of gyration, ρ feet.

It is usually assumed that all this weight is in the rim, or, in other words, the weight of the arms and hubs is neglected at this time, so far as their flywheel effect is concerned. The radius of gyration may be found by assuming a proper rim velocity. This should be kept below 6000 on large engines and below 5000 on small ones. After the weight is found, the rim dimensions may be computed from the formula

$$2 \times 12 \pi \rho \times a \times b \times 0.26 = W,$$

where b = width of rim,

a = thickness of rim, both in inches.

The ratio of $\frac{a}{b}$ should be assumed and used with above equation. After a has been found the outer and inner diameters will be found to be $2\rho + a$ for the outer and $2\rho - a$ for the inner.

Wheels that are to serve as belt-wheels will naturally have wider and thinner rims than plain flywheels. The width of belt should be determined, then the width b made about 1 inch greater.

Coefficient of Fluctuation. — The value of k to be assumed in the expression $\frac{N_1 - N_2}{N} = \frac{1}{k}$ depends on the nature of the work the engine is to do. In general it might be said that engines that drive through belts do not require so large a flywheel as engines that are direct-connected to the machine they drive. The following values of k are taken from Levin's "Modern Gas Engine and the Gas Producer."

	k
For ordinary industrial purposes, belt-driven	25– 35
For small electric light installations, direct-current, belt-driven	50– 60
For pumping machinery, direct-connected to engine	60–100
For large electric light installations, direct-current, belt-driven	60– 80
For large electric light installations, direct-connected	90–120
For gear-wheel transmissions	90–120
For blast-engines	90–150

Haeder and Huskisson, in their "Hand Book on the Gas Engine" give the following values:

For direct-current, direct-connected running singly	120
For direct-current, direct-connected running in parallel	200

For alternating-current, direct-connected running singly...	200
For alternating-current, direct-connected running in parallel	250
For direct-current, belt-connected, running singly........	80
For direct-current, belt-connected, running in parallel.....	120
For alternating-current, belt-connected, running singly....	100
For alternating-current, belt-connected running in parallel.	180

Construction Details. — Flywheels up to 8 feet in diameter may be cast solid, but all above this diameter should be cast in halves or smaller sectors. The material is usually cast-iron. In cases where extremely high speed is necessary the wheel may be made of steel.

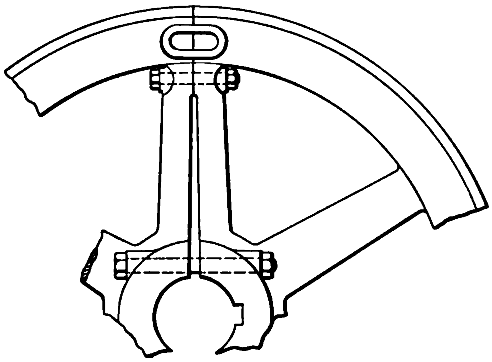


Fig. 187.

The simplest form of built-up wheel is the one cast in halves. In this case the parting should be made on an arm and not between the arms. Fig. 187 shows one manner of fastening the halves together. In addition to the hub bolts which clamp the

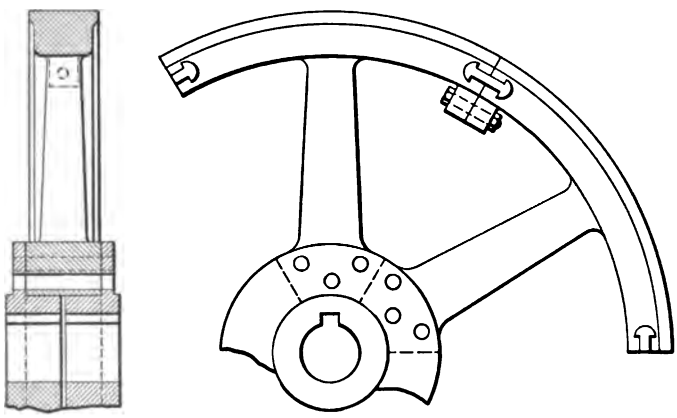


Fig. 188.

wheel on the shaft, there should be one or two keys. The total area of the bolts and shrink links at the rim should be at least one-fifth the sectional area of the rim.

When the wheel is built in sectors, usually six pieces, each arm is cast with a portion of the rim. The pieces of rims are held together by shrink links and bolts and the inner ends of the arms are bolted between two flanges at the hub. This method is illustrated in Fig. 188.

Tension in Rim. — The tension in the rim due to the centrifugal force may be found as follows:

$$\frac{awV^2}{gR} = \text{centrifugal force of 1 inch of rim,} \qquad (1)$$

when a = sectional area of rim in square inches,
w = weight of metal in pounds per cubic inch,
V = velocity of rim in feet per second,
R = maximum radius of rim in feet,
$R = \frac{D}{24}$, when D is the diameter in inches.

Then the centrifugal force becomes $\frac{24\,awV^2}{gD} = \frac{3\,awV^2}{4\,D}$.

The resultant force on one side of a diameter will be $D \times \frac{3\,awV^2}{4\,D}$ and this force will be resisted by $2\,Ka$, the tension in both sides of the rim. Then

$$2\,Ka = D \times \frac{3\,awV^2}{4\,D}, \qquad (2)$$

$$K = \frac{3\,wV^2}{8}. \qquad (3)$$

Design of the Arms. — Flywheel arms are usually made elliptical in section, with the major axis twice the minor. The usual number of arms is six. In very heavy flywheels eight arms are sometimes used.

The arms should be made strong enough to withstand the maximum twisting moment which it is possible for them to receive from the piston. This is approximately $p \times A \times r$, where

p = maximum pressure per square inch on piston,
A = area piston in square inches,
r = crank radius in inches.

This twisting moment will be equally distributed among all the arms, hence the bending moment on one arm will be

$$\frac{p \times A \times r}{n} \quad \text{where} \quad n = \text{number of arms.}$$

The section modulus of an ellipse is $\frac{\pi ab^2}{32}$, or $\frac{\pi b^3}{64}$, when a, the minor axis, is one-half the major. Then

$$\frac{p \times A \times r}{n} = \frac{K\pi b^3}{64}, \tag{1}$$

$$b = \sqrt[3]{\frac{P \times A \times r \times 64}{\pi n K}}. \tag{2}$$

The stress K may be assumed to be 2000 when used under the conditions mentioned above. The value of b found by (2) would be at the center of the shaft. The dimension b' at the rim may be found in several ways, one of which is to assume the taper to be $\frac{3}{8}$ inch per foot over all. Another way is to assume b' to be $\frac{3}{8}$ b. The minor axis a' at the rim may be found by either method. If the taper is used it should be limited to $\frac{3}{16}$ inch per foot. To find the dimensions of the arms at the hub when the drawing of the flywheel is made, use the taper as given above.

After the dimensions of the arms at the hub have been determined, the stress on this section, due to the centrifugal force of $\frac{1}{n}$th of the rim, should be determined. To this should be added the tension due to the weight of one-half the rim, since an arm in the bottom vertical position is assumed to carry the weight of half the rim. The total stress in an arm at the hub, when the flywheel is up to speed will be $K_c + K_w$, where

K_c = stress due to centrifugal force of rim,
K_w = stress due to weight of one-half the rim.

Example. — Design the flywheel arms for a 16 × 24-inch gas-engine, maximum pressure in the cylinder 400 pounds per square inch. Weight of rim 10,600 pounds, diameter 9 feet. Velocity of rim 5600 feet per minute.

Assume 6 arms.

$$M_b = \frac{400 \times 201 \times 12}{6} = 160{,}800 \text{ inch-pounds},$$

from relation on page 327.

From (2) we get

$$b = \sqrt[3]{\frac{160{,}800 \times 64}{\pi \times 2000}} = 11\tfrac{7}{8} \text{ inches}.$$

If we assume the radius out to the junction of the arms to be

1 foot, the width of section there will be $11\frac{1}{2}$ inches, thickness $5\frac{3}{4}$ inches.

The centrifugal force of one-sixth of the rim will be

$$\frac{10{,}600}{6} \times \frac{5600 \times 5600}{3600 \times 32 \times 4.5} = 109{,}000 \text{ pounds.}$$

$$K_c = \frac{109{,}000}{\frac{\pi}{4} \times 11.5 \times 5.75} = 2092 \text{ pounds per square inch.}$$

$$K_w = \frac{10{,}600}{2 \times \frac{\pi}{4} \times 11.5 \times 5.75} = 102 \text{ pounds per square inch.}$$

Total direct tension $= 2092 + 102 = 2194$ pounds per square inch. This stress should not exceed 2500 pounds per square inch.

Stress in Arms Due to Inertia of Rim. — The design of the flywheel arms as given in a previous article takes into account the total piston pressure and hence will give a certain size of flywheel arms for a given size of engine. In other words, the size of the arms will be independent of the size of the rim. In some cases arms designed in the manner given above will be weak. In any case where a large load is thrown on the engine suddenly, slowing the engine down, the inertia of the flywheel tends to keep the engine up to speed and the force of the inertia bends the flywheel arms. One very good illustration is the engine driving a rolling-mill. As the bar of steel strikes the rolls the load on the engine is increased instantly from zero to a maximum. The flywheel then keeps the engine going until the governor opens the valves. Gas-engines are rarely used on rolling mills but the same effect might be secured in an engine driving plunger pumps, or even in engines driving electric generators.

The arms of a flywheel should be tested to see what stress will result when the engine stops quickly. The weak point in this step is that the time of stopping must be assumed. After the time is assumed the force necessary to stop the wheel in that time may be found from $F = Ma = \frac{Wa}{g}$, where F = force in pounds, W = weight of rim in pounds and a = negative acceleration in feet per second per second. The force F will be applied out at the center of gravity of the rim and will bend the arms at the hub with a moment Fl, where l is the distance from the center of gravity of the rim to the hub.

Example. — The flywheel used in the last example had a rim 9 feet in diameter, weight of rim 10,600 pounds, velocity of rim 5600 feet per minute or 93.3 feet per second. This velocity corresponds to 200 r.p.m. If we assume the flywheel to make two revolutions in coming to rest, the time required will be

$$\frac{2 \times 9 \times \pi \times 2}{93.3} = 1.2 \text{ seconds.}$$

The force of acceleration of the rim will then be

$$F = \frac{10{,}600 \times 93.3}{32.2 \times 1.2} = 25{,}600 \text{ pounds.}$$

We have already made the assumption of the distance from the center of the wheel to the junction of the arms to be 12 inches. If we make the further assumption that the rim is 4 inches thick, the lever arm of the force F will be 40 inches. The bending moment on one arm will then be

$$M = \frac{25{,}600 \times 40}{6} = 170{,}500 \text{ inch-pounds,}$$

and the stress will be

$$K = \frac{170{,}500 \times 64}{\overline{11\frac{1}{2}}^3 \times \pi} = 2280 \text{ pounds per square inch.}$$

Velocity and Displacement Diagrams.—Let it be assumed that the length of the rotative effort diagram for the single-cylinder engine, Fig. 185, represents the time it takes the flywheel to make two revolutions instead of representing the distance traveled by the crank-pin in one cycle or two revolutions. This assumption may be made without error because the difference between the intervals of time and intervals of space is so small that it could not be measured on the diagram.

The driving effort acting tangentially on the crank-pin is represented for all crank positions by the ordinates of the crank-effort curve, but by multiplying this force, measured at any point of the cycle, by the quantity $\frac{g}{W}$ we obtain the acceleration which the tangential effort gives a weight W. The tangential effort curve can, therefore, be said to be the acceleration curve for a weight W.

The distance between successive ordinates of the diagram represents the time required for $\frac{1}{24}$ of a revolution. The height of the

mean ordinate between the base-line and the acceleration curve during any interval represents the average of the variable acceleration which a weight W is given during the time $\frac{1 \times 60}{24 \times N}$ seconds, $\frac{60}{N}$ being the time for one revolution.

The change in the velocity of a moving object during a given period of time is the product of its acceleration and the time. Hence, the area of the curve, Fig. 185, which we may now call the acceleration curve, up to any given ordinate, represents the velocity at that point or, in other words, the area included between any two ordinates, the curve and the mean effort line represents the change in velocity during the time represented by the distance between the ordinates.

In order to represent graphically the increase or decrease in the velocity of the revolving parts of the engine we may, therefore, integrate the elementary areas beginning at any point of the cycle and plot their sum on corresponding ordinates, having as a base-line the base of the acceleration curve. In Fig. 189 is shown

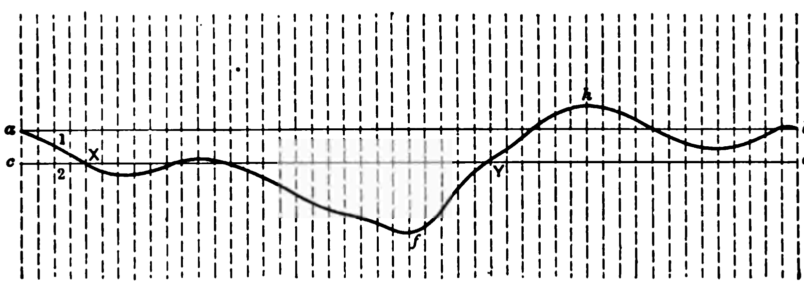


FIG. 189. — Velocity Diagram for a Four-cycle, Single-cylinder, Single-acting Gas-engine.

such a velocity curve plotted from the rotative effort or acceleration curve in Fig. 185. The velocity curve was started at the point on the acceleration curve corresponding to the beginning of the suction stroke.

The line *ab* on Fig. 189 is the base-line and *cd* the mean velocity line, the distance between *ab* and *cd* being the area between the curve and base-line divided by the length of the curve.

The vertical distance between the low point of the curve *f* and the high point *h* represents the maximum change in velocity of

the flywheel during the time represented by the horizontal distance between f and h. During this time the flywheel absorbs or gives out the energy represented by ΔE in Fig. 185.

The area between the line cd, the **velocity curve** and any two ordinates is the product of the excess or deficiency in velocity and the time. Hence we may integrate the areas curve and cd, the line of normal velocity, and the area thus found up to any point will be proportional to the displacement of a point on the flywheel at that time. If the velocity of the revolving system is constant, then each point of the system will be in its proper position each instant, but as the velocity varies from the normal, so will the position of a given point vary from the normal and the variation is shown by the ordinates of Fig. 190, which is the **displacement curve.**

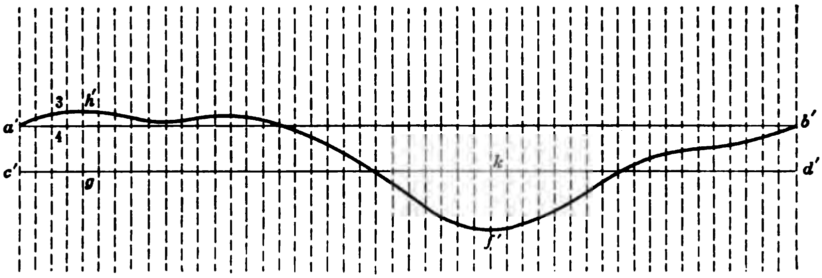


FIG. 190. — Displacement Diagram for a Four-cycle, Single-cylinder, Single-acting Gas-engine.

The line $a'b'$ of Fig. 190 is the base-line from which the ordinates are set off, representing areas of the velocity curve up to the corresponding ordinate on the velocity curve. Thus, the ordinate 3–4 of the displacement curve is proportional to the area a 12 c of the velocity curve and is laid off in the positive direction since the area a 12 c is above the mean line and hence positive.

Since a point in the revolving system will continue to fall further behind its normal position so long as the velocity is below normal, it follows that the point of maximum displacement ahead of normal will be at the end of a period of velocity above the normal. Thus the point where the velocity curve crosses the mean velocity line at X is the end of a positive velocity area. Hence the maximum ahead displacement h' will occur directly under X on the velocity curve. Likewise the maximum displacement in the oppo-

site direction will occur at f', directly under Y, the point at which the velocity curve crosses the mean velocity line after a period of low velocity.

The area of the velocity curve above the mean velocity line is equal to the area below the mean velocity line. If the velocity were constant, then there would be no displacement ahead or behind the normal. Since the displacement is proportional to the area of the velocity curve, it follows that a revolving point will get a certain distance ahead of its normal position during a revolution and will fall the same distance behind, but at some point, or at two points in the revolution, it will be in its normal position. Hence, if we draw a horizontal line $c'd'$, Fig. 190, in such a way that gh' equals $f'k$, then the ordinates from this line to the curve will be proportional to the distance the flywheel is ahead of or behind its normal position at the time corresponding to the position of the ordinate.

Application of the Velocity and Displacement Diagrams. — Let us assume a condition which places an alternating-current generator driven by a water-wheel in parallel operation with one driven by a gas-engine, the generator in each case to be direct-connected to its prime mover. Let us also assume the power taken from the generators during a period of time corresponding to a revolution to be constant. Then the generator driven by the water-wheel will revolve at a constant velocity but the one driven by the gas-engine will get ahead of and lag behind the other an amount proportional to the maximum displacements as shown by the displacement diagram. The position of the gas-engine-driven generator at any time relative to the water-wheel-driven generator is proportional to the ordinate of the displacement curve at that instant.

The actual amount which the gas-engine-driven generator gets behind or ahead of the other depends on the combined flywheel effect of the flywheel and generator. The flywheel formula may be written

$$W = \frac{2\,g\Delta E}{2\,V\,(V_1 - V_2)} = \frac{2\,g\Delta E}{4\,\pi\rho N\,(V_1 - V_2)},$$

where W is the combined weight of flywheel and rotor reduced to a common radius of gyration ρ. The weight increases as $(V_1 - V_2)$ decreases, and consequently as the displacement decreases. The weight, then, is dependent on the allowable dis-

placement. This allowable displacement is usually given as $2\frac{1}{2}$ degrees phase, or it is $\frac{2.5}{360} = 0.0069$ of the distance from one positive pole to the next positive pole. If the generator is a two-pole machine, then the displacement on the flywheel rim will be $2\frac{1}{2}$ degrees. If n = the number of poles on the generator, $\frac{2.5 \times 2}{n}$ will be the degrees displacement on the flywheel circle. If given in radians the angle will be $\frac{2.5 \times 2 \times \pi}{180\,n}$. The relation between this angle and the displacement curve is given here.

In Fig. 185 it is evident that the part above the mean line marked ΔE is the excess energy. Let F be the height of the curve above the mean line at any point within the area ΔE. Then the excess torque at this point is

$$FRAS = \frac{W\rho^2}{g}\frac{d^2\theta}{dt^2}, \tag{1}$$

where R = radius of crank in feet,
A = area piston in square inches,
F = height of curve above mean line in inches at any point within the area ΔE.
S = scale of curve, pounds per inch,
W = weight of flywheel and generator rotor reduced to ρ feet,
where ρ = radius of gyration.

If we multiply both sides of equation (1) by dt and change $\frac{d^2\theta}{dt^2}$ we get

$$FRAS\,dt = \frac{W\rho^2}{g}\frac{d}{dt}\left(\frac{d\theta}{dt}\right)dt. \tag{2}$$

Since the length of the curve in Fig. 185 corresponds to 2 revolutions, we have for the time in seconds, per inch of curve,

$$t = \frac{2 \times 60}{lN}, \tag{3}$$

where l = length of curve in inches and N = r.p.m. The time dt required for a differential length dx will be

$$dt = \frac{2 \times 60}{lN}dx. \tag{4}$$

Substituting this value of dt in (2) and integrating,

$$\frac{RASg}{W\rho^2}\frac{120}{lN}\int F\,dx = \int\frac{d}{dt}\frac{d\theta}{dt}\,dt = \frac{d\theta}{dt}. \tag{5}$$

Let $\frac{RASg}{W\rho^2}\frac{120}{lN} = C.$ $\int F\,dx$ is the area enclosed by the rotative effort curve and the mean line up to any point and $\frac{d\theta}{dt}$ is the angular velocity of that flywheel at that point. When $\int F\,dx$ up to any point is multiplied by C, the product will be the angular velocity of the flywheel at that instant in radians per second. Then (5) becomes

$$C \times \text{area acceleration curve to any point} = \frac{d\theta}{dt} = \omega. \qquad (6)$$

When the velocity curve is constructed from the acceleration curve, a scale must be selected. Thus, 1 inch on the ordinates of the velocity curve will represent a certain number of square inches of the area of the acceleration curve. Call this scale S_1. Let h_1 = the height of the velocity diagram above or below the mean line at any point. Then in (6) we may substitute $h_1 \times S_1$ for the area of the acceleration curve to any point. If that is done and if both sides of the equation are multiplied by dt and the equation integrated, the result will be

$$C \times S_1 \int h_1\,dt = \int \frac{d\theta}{dt}\,dt. \qquad (7)$$

Substituting for dt its value from (4), we get

$$C \times S_1 \times \frac{120}{lN}\int h_1\,dx = \theta, \qquad (8)$$

where θ is the angular displacement in radians, since $\int h_1\,dx$ is the area of the velocity curve up to any point.

If the scale on the displacement curve is S_2 square inches of the velocity curve per inch of height h_2, then for $\int h_1\,dx$ in (8) we may put S_2h_2 giving

$$CS_1S_2 \times \frac{120}{lN} \times h_2 = \theta. \qquad (9)$$

Equation (9) may be used to find the weight of the flywheel for a given value of θ or it may be used to find the angular displacement θ for a flywheel determined by some other method. If the first method is used the value of θ should be found from the allowable fluctuation in degrees of phase. Thus, if the allowable

fluctuation is α degrees of phase and n is the number of poles on the alternator,

$$\theta = \frac{\alpha \times 2 \times \pi}{180\, n}. \tag{10}$$

If equation (9) is used to find the displacement with a flywheel of a given weight, the value of θ thus found should be less than that indicated by equation (10).

Approximate Weight of Flywheel. — It is sometimes desirable to find the approximate weight of a flywheel without finding the rotative effort diagram for a given engine. The excess energy ΔE which must be absorbed and given out by the flywheel bears some relation to the work generated per revolution or per cycle in the engine cylinder. This ratio

$$\frac{\text{Energy absorbed by flywheel}}{\text{Indicated work per revolution}}$$

is practically constant for a given type of engine.

The author has found the following values for the ratio given above:

TABLE XVI

FLYWHEEL CONSTANTS FOR APPROXIMATE CALCULATION

Type of engine, four-cycle.	ΔE ÷ indicated work per revolution.
Single-acting:	
Single-cylinder	2.35–2.40
2-cylinder twin, cranks at 360 degrees	0.92–1.04
2-cylinder twin, cranks at 180 degrees	1.50–1.60
2-cylinder opposed, cranks at 360 degrees	1.50–1.60
2-cylinder opposed, cranks at 180 degrees	0.92–1.04
2-cylinder tandem	0.92–1.04
3-cylinder vertical, cranks at 120 degrees	0.60–0.75
4-cylinder vertical, crank at 90 degrees	0.18–0.20
Double-acting:	
Single-cylinder	1.50–1.60
2-cylinder tandem	0.18–0.20
2-cylinder twin	0.18–0.20
2-cylinder twin-tandem, cranks at 90 degrees	0.07–0.075

Example. — Find the weight of flywheel for a 1000-I.H.P. 2-cylinder tandem double-acting gas-engine running 85 r.p.m. Assume all the weight to be in the rim. Coefficient of fluctuation is $\frac{1}{150}$. Diameter of wheel 20 feet.

$$\Delta E = \frac{0.2 \times 1000 \times 33{,}000}{85} = 78{,}000 \text{ foot-pounds.}$$

From equation (4), page 324,

$$W = \frac{\Delta E \times g \times 900 \times k}{\pi^2 \rho^2 N^2}$$

$$= \frac{78{,}000 \times 32 \times 900 \times 120}{\pi^2 \times 10 \times 10 \times 85 \times 85} = 37{,}300 \text{ pounds.}$$

EXERCISES

1. A cast-iron flywheel rim weighs 18,000 lbs. and is 12 ft. mean diameter. Find dimensions of rim if section of rim is square.

2. Given: $\Delta E = 43{,}000$ ft.-lbs., radius of gyration 8 ft., r.p.m. 150. Find weight of flywheel rim for a gas-engine direct-connected to a direct-current dynamo running singly.

3. Find size of arms for a flywheel 16 ft. dia. for a 24×32-in. engine, explosion pressure 400 lbs. Outer radius of flywheel hub 12 ins. Use six arms of elliptical section, major axis of ellipse twice the minor.

4. A flywheel rim weighs 28,600 lbs. and has six arms of elliptical section, $13\frac{1}{2} \times 6\frac{3}{4}$ ins. Distance from center of gravity of rim to hub 4′–3″. Find stress in arms if flywheel comes to rest in two revolutions, from speed of 175 r.p.m.

5. Find weight of flywheel rim for 75 horse-power single-cylinder, single-acting, four-cycle engine running 175 r.p.m., wheel to have radius of gyration of 4 ft. Assume $k = 60$.

6. Design the arms for flywheel in Prob. 5. Find total weight of wheel.

CHAPTER XXII

CYLINDERS — CYLINDER COVERS — FRAMES — VALVES — VALVE-GEARS

Cylinders. — The **materials** used in gas-engine cylinders are cast-iron and cast-steel. If iron is used it should be hard and close-grained. Steel is used only where cast-iron liners are used. Allowable stress is fairly high so that the walls may be thinner than if a lower stress is used. With thinner walls conduction of heat is more rapid than with thick walls and stresses due to heating up are lower. For cast-iron with clearly defined load conditions, the allowable stress in tension may run as high as 4000 pounds per square inch.

Cylinders and jackets are often cast in one piece and in very small engines the cylinder, jacket and frame may be combined in one casting, although this is not the usual practice. In some instances where the cylinder and jacket are cast as one piece, they are connected at one end only and are free from each other at the other end. This arrangement is shown clearly in Fig. 191 which illustrates the cylinder of a vertical Diesel engine. The liner and cylinder are connected at the top but left free at the bottom where the opening thus formed in the water-space is filled by a ring held in place by means of cap-screws. The same method is sometimes used in double-acting cylinders. In this case the jacket is cast integral with the cylinder but is not continuous from end to end. A portion is left out at the middle of the cylinder and the opening thus formed is closed by a separate steel band. Rubber rings are laid in U-shaped grooves at the ends of the jacket and the bands are pulled up on these rings, forming a water-tight joint. This method is illustrated in Fig. 192. Figs. 67 and 92 illustrate clearly the practice of casting the cylinder and jackets in one piece without leaving the jacket free at one end. Fig. 108 illustrates an engine in which a separate liner has been used, the liner forming the cylinder proper and the space between the liner and jacket the water-space. A separate liner has the advantage that a specially suitable grade of iron can be employed and the jacket

and frame may be cast with greater ease. It has the additional advantage that the wear on the cylinder may be taken care of

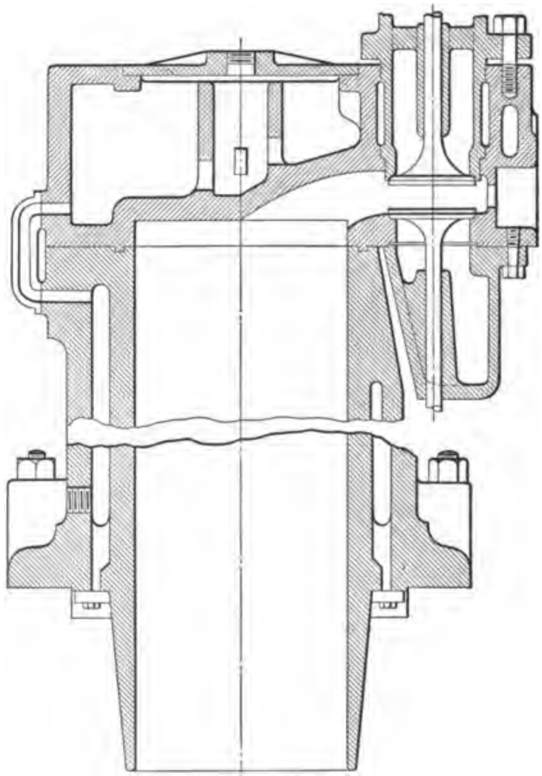


FIG. 191. — Cylinder for Vertical Multicylinder Gas-engine or Diesel Engine.

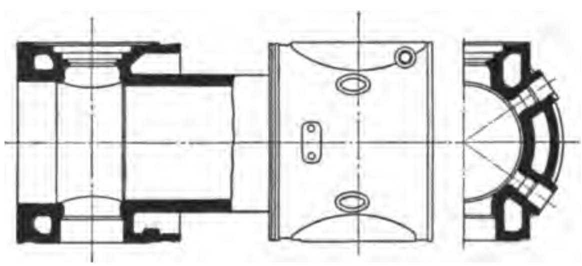


FIG. 192. — Double-acting Gas-engine Cylinder with Non-continuous Jacket.

more readily. A further advantage, one that is very important, is that expansion due to high temperature in the cylinder is taken

care of without imposing dangerous stresses on cylinder and jacket. To allow this free expansion, the liner is held rigidly at one end while the other is allowed to move axially.

Constructive Details. — The thickness of the cylinder should be determined to withstand the internal pressure of the explosion

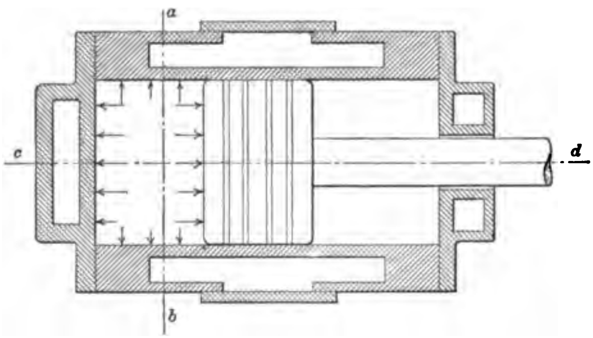


FIG. 193.

and the stresses brought on it by the connecting-rod thrust in a trunk-piston engine. The internal pressure stresses the cylinder barrel in a longitudinal direction and in a radial direction as shown by Fig. 193. The cylinder wall proper takes all the radial load and all the axial load if the water jacket is not continuous and cast integral with the cylinder.

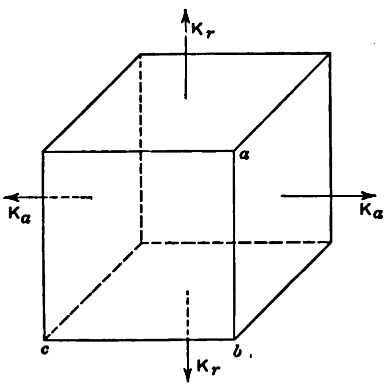


FIG. 194.

Let K_a be the unit stress due to the axial load and K_r due to the radial load. Then a unit cube of metal in the cylinder wall will be stressed as shown in Fig. 194. The stress K_r tends to increase the length of *ab* and decrease *cb*, while K_a has the opposite effect, tending to increase *cb* and decrease *ab*. From the stress strain relation we have, in general,

$$\frac{K}{e} = E, \quad \text{and} \quad K = Ee, \tag{1}$$

where K is the unit stress, e the unit strain or deformation and E

is the modulus of elasticity. The relation between the elongation of ab and compression of cb due to K_r is given by $\frac{1}{m}$, Poisson's ratio. That is, the decrease in the length cb due to K_r is $\frac{1}{m}\frac{K_r}{E}$ and the decrease in the length ab due to K_a is $\frac{1}{m}\frac{K_a}{E}$. Hence the net deformation of ab is

$$e_1 = \frac{K_r}{E} - \frac{1}{m}\frac{K_a}{E}. \tag{2}$$

Using (2) and (1) we may find the net stress K_1 due to the net deformation e_1.

$$K_1 = Ee_1 = K_r - \frac{1}{m}K_a. \tag{3}$$

Likewise the stress in direction of cb is

$$K_2 = Ee_2 = K_a - \frac{1}{m}K_r. \tag{4}$$

The thickness of the cylinder wall may be found by using the formula for a thin cylinder, neglecting the stress due to the pressure on the ends of the cylinder,

$$t = \frac{pD}{2\,K_r} + C, \tag{5}$$

where p is the internal pressure in pounds per square inch, D the diameter in inches, K_r the allowable fibre stress and C a constant to allow for reboring. The constant depends on the diameter of the cylinder and may be selected from Table XVII.

TABLE XVII

VALUES OF C

Diameter of cylinder...............	4	6	8	10	12	14	16	18	20	24	28	32	40	48
C in inches.........................	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{13}{16}$

Example. — Find the thickness of cylinder walls for a 12-inch gas-engine cylinder, explosion pressure 450 pounds per square inch.

$$t = \frac{450 \times 12}{2 \times 4000} + \frac{1}{2} = 1\tfrac{3}{16} \text{ inches.}$$

The stress due to the pressure on the ends of the cylinder may be found by the formula

$$\frac{\pi}{4}(D_1^2 - D^2)\,K_a = \frac{\pi D^2}{4}p, \tag{6}$$

or
$$K_a = \frac{D^2}{D_1^2 - D^2} p, \tag{7}$$

where D is the inside and D_1 the outside diameter of the cylinder in inches. In the problem above $D = 12$ inches and $D_1 = 14\frac{3}{8}$ inches. Hence

$$K_a = \frac{\overline{12}^2}{\overline{14.375}^2 - \overline{12}^2} \times 450 = 1035 \text{ pounds.}$$

The stress due to radial pressure will be

$$K_r = \frac{450 \times 12}{2 \times 1.18} = 2290 \text{ pounds.}$$

From (3) we find the stress due to the combination of K_a and K_r to be

$$K = 2290 - \frac{1}{4} \times 1035 = 2030 \text{ pounds,}$$

where $\frac{1}{4} = \frac{1}{m}$ for cast-iron. Since the resulting stress is almost as large as the maximum stress, the maximum is used in cylinder design. Concerning the above problem the stresses will increase greatly after the cylinder is rebored and care should be taken that reboring does not increase the stresses until they become dangerously high.

Some designers of gas-engines use the formula for thick cylinders for finding the thickness of the cylinder wall. The one used by Güldner in his "Design and Construction of Internal Combustion Engines" is

$$t = \frac{D}{2}\left(\sqrt{\frac{K + 0.4\,p}{K - 1.3\,p}} - 1\right) + C \text{ inches.} \tag{8}$$

Using the same example as before we find

$$t = \frac{12}{2}\left(\sqrt{\frac{4000 + 0.4 \times 450}{4000 - 1.3 \times 450}} - 1\right) + \frac{1}{2} = 1.142 \text{ inches.}$$

Due to the internal pressure the inner fibres of the cylinder wall are under a greater stress than the outer fibres. This inequality of stress increases materially with an increase in the thickness of the wall. For that reason t should not be made greater than is necessary. Excessive thickness is also wrong from the standpoint that the cooling water is less effective, consequently the conduction of heat is less rapid, and troublesome overheating of the material may ensue.

Stress in Cylinder Walls Due to Connecting-rod Thrust. — In large or long-stroke engines the thrust of the connecting-rod should receive careful attention in finding the thickness of the cylinder. In case the formulæ given should furnish a section modulus too small to take care of the normal pressure T due to the connecting-rod, it is better to increase the stiffness of the cylinder wall by suitable supports or webbing rather than to increase the thickness.

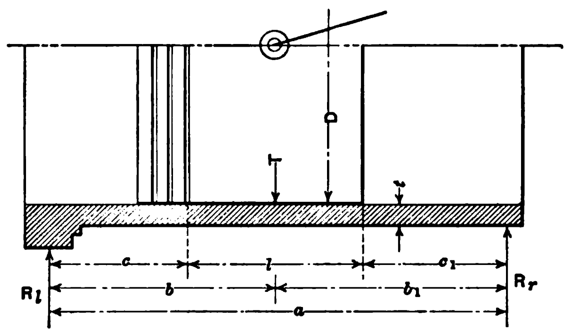


FIG. 195.

With the favorable assumption that T is distributed over the bearing surface of the length l, Fig. 195, the maximum bending moment will be

$$M = R_l\left(c + \frac{lR_l}{2\,T}\right) \leqq \frac{K_b I}{d_1}, \tag{1}$$

in which

$$R_l = \frac{T\,(2\,c_1 + l)}{2\,a} \tag{2}$$

and

$$\frac{I}{d} = \text{section modulus.}$$

The stiffness of the cylinder liner or the cylinder should be such that the vertical component T cannot cause any noticeable bending or deformation. In order to make sure of this point it is advisable to make a check computation, assuming that the force T, instead of being distributed over the length l, is concentrated at the middle of this length. In that case

$$M = T\frac{bb_1}{a} \text{ inch-pounds,} \tag{3}$$

and assuming that K_b is the allowable stress, the section modulus required will be

$$\frac{I}{d_1} = T\frac{bb_1}{aK_b} \geqq \frac{M}{K_b}. \tag{4}$$

The deflection of the cylinder will be

$$\delta = \frac{Tb^2b_1^2}{3\,aIE}.$$

Example. — A cylinder liner with a diameter of 25 inches and an explosion pressure of 400 pounds per square inch will have a thickness of

$$t = \frac{400 \times 25}{2 \times 4000} + \frac{9}{16} = 1\tfrac{13}{16} \text{ inches.}$$

With reference to Fig. 195 let $a = 50$ inches, $b = 24$ inches, $b_1 = 26$ inches. Then the maximum value of T will be approximately

$$T = 0.10 \times \frac{\pi \times 25^2}{4} \times 400 = 19{,}640 \text{ pounds.}$$

Then $$M_b = \frac{19{,}640 \times 24 \times 26}{50} = 245{,}000 \text{ inch-pounds.}$$

$$\frac{I}{d} = \frac{\pi}{4}D_m^2t = 0.7854 \times \overline{26.81}^2 \times 1.81 = 1023,$$

where D_m is the mean diameter of the cylinder barrel. The maximum bending stress therefore is

$$K_b = \frac{245{,}000}{1023} = 240 \text{ pounds per square inch.}$$

This stress is low and apparently the cylinder is amply strong but the maximum deflection should be determined. The moment of inertia of the cross-section is

$$I = \frac{\pi D_m^3t}{8} = \frac{\pi \times \overline{26.81}^3 \times 1.81}{8} = 13{,}700.$$

Using a value of 12,000,000 for E for cast-iron we find the maximum deflection to be

$$\delta = \frac{19{,}640 \times \overline{24}^2 \times \overline{26}^2}{3 \times 50 \times 13{,}700 \times 12{,}000{,}000} = 0.000315 \text{ inch.}$$

Since the load T is not concentrated but is distributed over the length l which makes δ still smaller, the result shows that this cylinder liner is safe without any further stiffening.

Cylinder Flanges. — In many cases gas-engine cylinders do not have flanges but the studs which retain the head in place are screwed into the metal joining the cylinder wall and jacket wall. This condition is clearly shown in Figs. 191 and 192. In cases where it is necessary to use a flange this flange should be made thicker than the cylinder wall and the depth of the flange *a*, Fig. 196, should be made as small as possible, just large enough to accommodate the bolts. The stress on the bolts due to drawing up the packing at *b* combined with the explosion pressure tends to break the flange along the line *bd*. This necessitates making t' about 1.25 t, where t is the thickness of the cylinder wall, and also keeping a as small as possible. In addition a generous fillet should be used at the junction of the flange and cylinder wall.

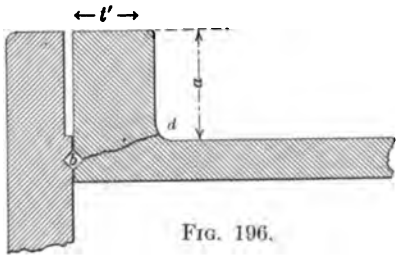


FIG. 196.

Jacket Wall. — The stress on the jacket wall depends entirely on the arrangement of cylinder and jacket walls. When the cylinder and jacket are cast in one piece, as in Fig. 197a, the jacket

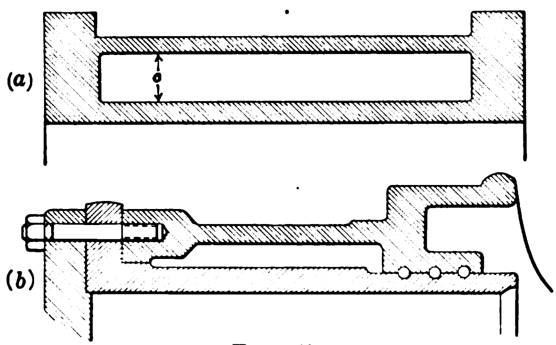


FIG. 197.

takes part of the axial thrust due to the explosion pressure. If the arrangement is as shown in Fig. 197b the jacket takes all the axial thrust and should be designed accordingly. In small engines sheet-iron or copper jackets are sometimes used. Where they are used the cylinder wall must be depended on to take all loads due to the explosion. In general it might be said that the thickness of the jacket wall should be from two-thirds to three-

fourths of the thickness of the cylinder wall. The depth of the water space c, Fig. 197a, should be from $\frac{3}{4}$ inch for a 6-inch cylinder to 3 inches for a 30-inch cylinder.

Cylinder Cover. — The cylinder cover is usually made of tough, close-grained cast-iron. The allowable stress should, in such places where it can be determined with any accuracy, be kept moderately high, say 3500 pounds per square inch. This is done in order to get a construction that is not too rigid against expansion by heat and to obtain better conduction of heat through the walls. In general, however, the less the certainty with which the forces acting can be determined, the lower should be the value of the stress.

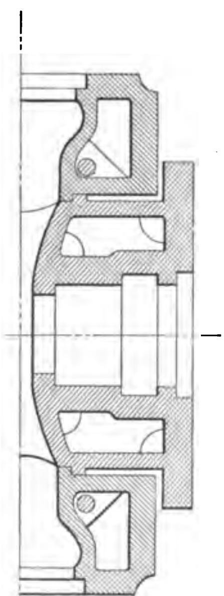


FIG. 198. — Cylinder Cover for a Double-acting Gas-engine.

In cases where the cylinder cover is a flat plate the thickness could be found from the formula

$$t = D\sqrt{\frac{cp}{K}},$$

where t = thickness in inches,
D = diameter of cylinder in inches,
p = pressure in pounds per square inch,
c is a constant and K the allowable fibre stress. The value of the constant is about 0.10 for gas-engine work.

Example. — Find the thickness of a plain cylinder cover for a 12-inch cylinder, pressure 450 pounds.

$$t = 12\sqrt{\frac{0.1 \times 450}{3500}} = 1\tfrac{1}{4} \text{ inches.}$$

This same example was used to find the thickness of the cylinder wall, which was found to be $1\frac{3}{16}$ inches. The usual steam-engine practice is to make the cylinder cover the same thickness as the cylinder walls. In the usual type of gas-engine the cylinder cover or cylinder head, as it is sometimes called, is very irregular in shape on account of the water jacket and valve chambers. Hence the metal may be made thinner as the section of the cover is more in the nature of a girder. This is shown in Figs. 191, 198 and 198a.

Frames. — The material ordinarily used for frames of stationary engines is cast-iron although occasionally steel is used. For automobile motors aluminum is frequently used. The allowable fibre stress is low in consideration of the importance of this machine part and because only slight deflections are permissible.

FIG. 198a. — Cylinder Cover for a Diesel Engine.

The body of vertical engines is usually called the frame, that of horizontal engines the bed, although the use of these terms is not at all strictly defined. Both of these forms have been fairly well standardized and these standards are closely adhered to by different manufacturers unless necessity demands other construction.

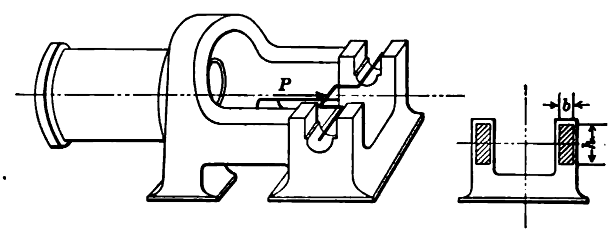


FIG. 199.

The simplest type of frame for the single-acting horizontal engine is shown in Fig. 199. In this type the neutral axis of the two beams connecting the sides of the frame to the bearings and cylinder is in the same horizontal plane as the center of the shaft.

Then these two beams are in direct tension and the stress in them is

$$K_t = \frac{P}{2a},$$

where P = total load on piston due to explosion pressure and a is the area of one of the side-beams = bh square inches. The stress K_t should not exceed 1500 pounds per square inch.

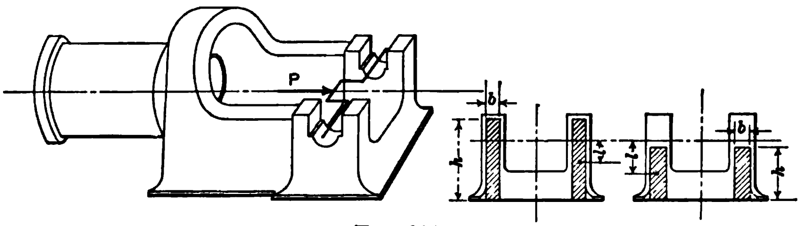


FIG. 200.

The frame illustrated in Fig. 200 is of a type common in small, single-acting engines. The force of the explosion P puts the two side members in tension and bending. The stress in tension will be

$$K_t = \frac{P}{2a},$$

as before. In addition there will be a stress due to the bending moment

$$M = Pl,$$

where l is the distance from the center-line of the engine to the center of gravity of the sides of the frame. The moment of inertia of the frame is $\frac{2\,bh^3}{12}$ and the section modulus $\frac{bh^2}{3}$. The bending stress is then

$$K_b = \frac{3\,Pl}{bh^2}.$$

The total stress in the frame will be the sum of the tension and bending stresses, or

$$K_{\text{total}} = K_t + K_b.$$

In medium-size single-acting engines the frame usually has a horizontal member as shown in Fig. 201. In this section the distance l from the center-line of the engine to the center of gravity of the section may be found after x is found.

$$x = \frac{2\,bh \times \frac{h}{2} + fc \times g}{2\,bh + fc}:$$

The moment of inertia of the section is then

$$I = \frac{2\,bh^3}{12} + 2\,bh \times e_1^2 + \frac{cf^3}{12} + cf \times e_2^2.$$

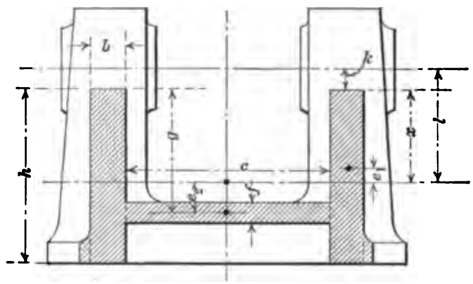


FIG. 201.

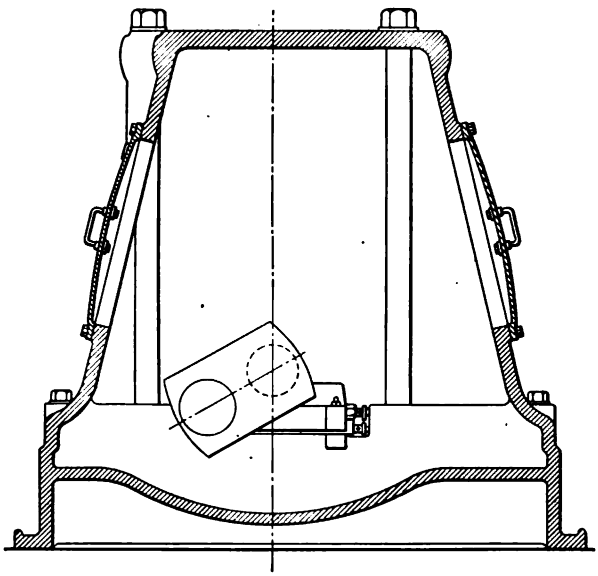


FIG. 202. — Frame for Single-acting Vertical Gas-engine.

The section modulus is $\frac{I}{x}$, and $l = k + x$. The bending stress in the top fibre is then

$$K_b = \frac{Pl}{\frac{I}{x}}.$$

The direct tension is

$$K_t = \frac{P}{2\,bh + fc}.$$

The total stress is

$$K_{\text{total}} = K_b + K_t.$$

It will not be necessary to find the stress in the bottom fibres as they are in compression, due to bending, and tension, due to the direct load P. The net stress will be the difference of these two and will be less than the total stress on the top fibres.

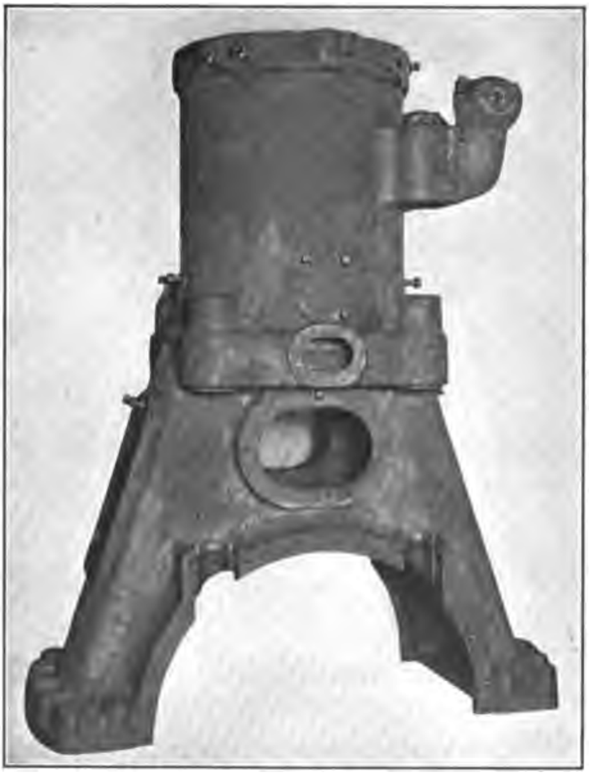


FIG. 203. — Frame and Cylinder for Single-acting Vertical Diesel Engine.

A frame for a vertical multicylinder engine is shown in Fig. 202. In this particular case long steel bolts are employed to take the tension due to the piston pressure. These bolts extend from the bottom of the frame to the plate on which the cylinders are bolted.

The section shown in the illustration was taken between two cylinders.

In Fig. 203 is illustrated a frame for a vertical Diesel engine with the cylinder jacket cast with the frame. The bosses for the tension bolts are shown at the corners. This frame is arranged to be bolted on to a cast-iron sub-base.

Large gas-engines usually have a frame which is entirely separate from the cylinder. The one illustrated in Fig. 204 is made by the Snow Steam Pump Works for their tandem engines. The crosshead guides are bored in the frame but the top guide may be removed to facilitate removal of the crosshead. This frame is for a side-crank engine. It will be noticed that the side of the frame opposite the bearing is made high to bring the center of gravity of the frame up near the center-line, thus keeping the bending stress low. Another method of keeping down the bending stress is to make the side members of the frame box-sections instead of solid webs.

It is impossible to use the common method applied to beams in designing engine frames. Instead of assuming a working stress and computing the dimensions it is necessary to assume the dimensions and compute the stress. This cut-and-try method will not prove successful on first trial; very often three or four trials are necessary to find a section that will be both safe and economical.

The bottoms of all frames should have flanges wide enough so that the unit bearing pressure due to weight of machine and whatever inertia forces there may be should not exceed the following figures for the various materials employed:

	Pounds per square inch.
Granite blocks	110–140
Sandstone	70– 80
Hard-pressed brick in cement	55– 70
Concrete	50– 65
Hardwood timber	55– 70
Pine and hemlock timber	30– 40

The attachment of the cylinder to the frame requires careful consideration on the part of the designer. The frame should be bored to receive a turned portion of the cylinder in order to have the two line up perfectly. In Fig. 205 is illustrated a cylinder

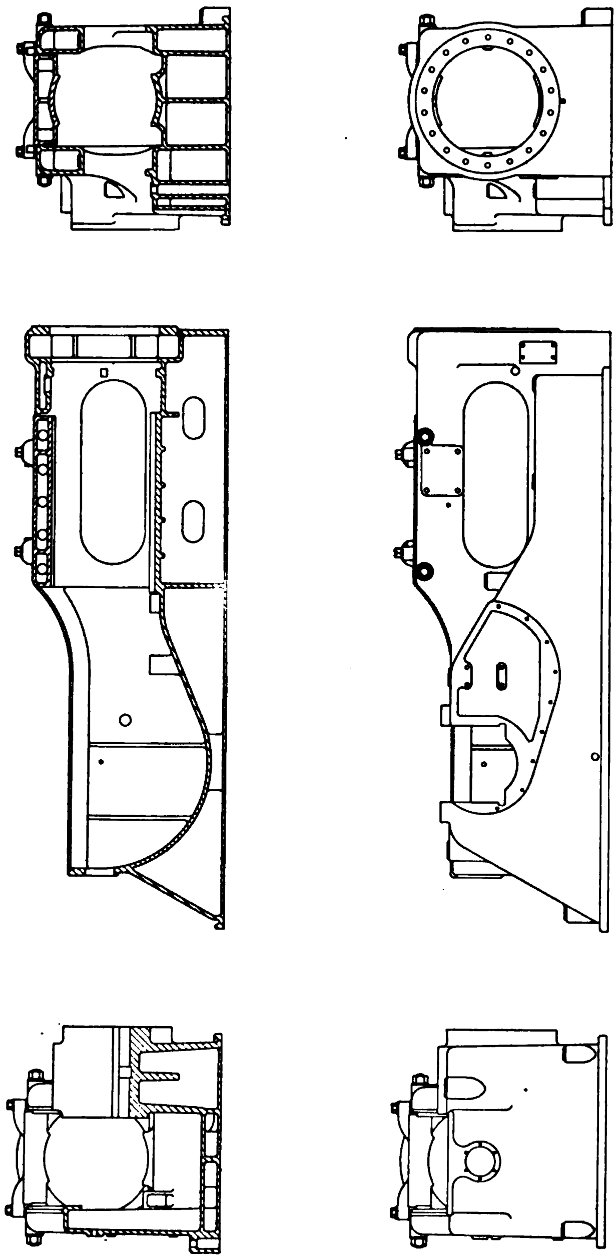


FIG. 204. — Frame for Large Double-acting Gas-engine.

for a two-cycle oil-engine. The shoulder *A* was machined to fit into the frame. This shoulder was turned on the boring mill at the same time that the inside of the cylinder was bored.

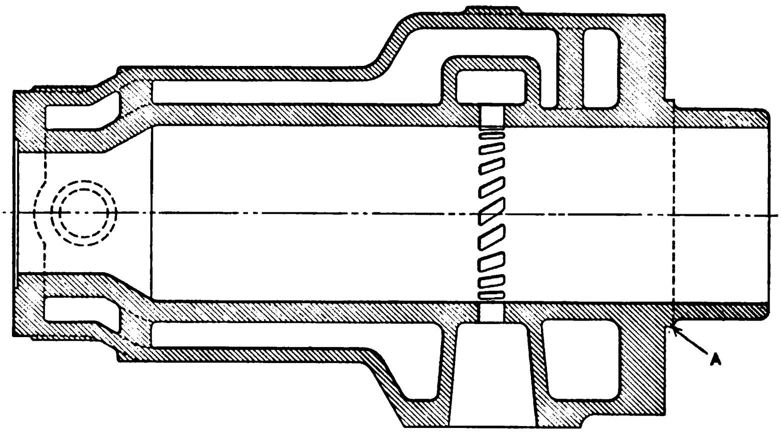


FIG. 205.

Figs. 206, 207 and 208 show details of **cylinder** and **frame connections.** The cylinder in Fig. 206 is cast integral with the jacket and studs are used for connection to the frame. Fig. 207

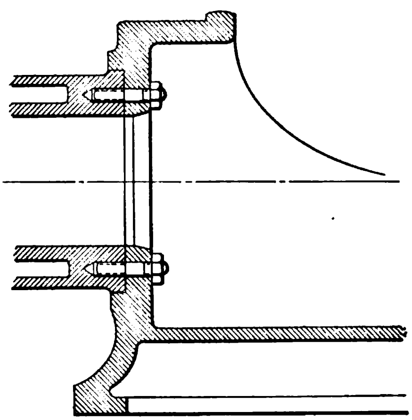


FIG. 206.

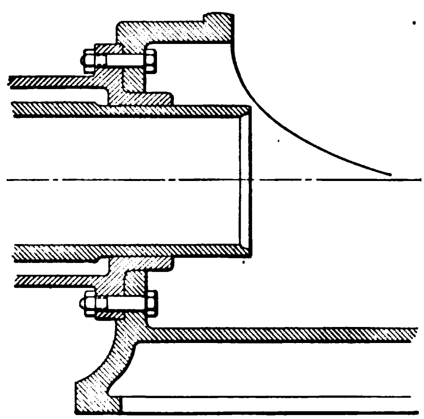


FIG. 207.

illustrates a cylinder with a separate water jacket. The water jacket is bolted to the frame by means of through bolts. The method of casting the jacket with the frame is illustrated in

Fig. 208. The construction used in Figs. 207 and 208 necessitates a water-tight joint where the cylinder extends through the frame. Figs. 209 and 210 illustrate two types of stuffing boxes used at this point. Any kind of soft packing may be used with the one shown in Fig. 209 but a rubber ring should be used with the one in Fig. 210.

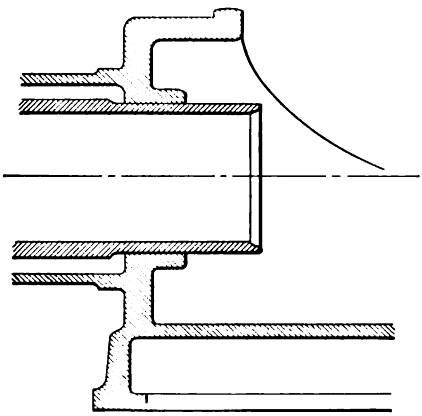


Fig. 208.

The **studs** or **bolts** used in fastening the cylinder head to the cylinder should be made of a good grade of soft steel or wrought iron. The diameter of the studs may be found by equating the total load on the head due to the explosion pressure to the area of all the studs at the root of the thread multiplied by the allowable fibre stress. Thus

$$\frac{\pi d^2}{4} \times p = \left(\frac{n\pi d_1^2}{4}\right) K_t \qquad (1)$$

and

$$d_1 = d\sqrt{\frac{p}{nK_t}}, \qquad (2)$$

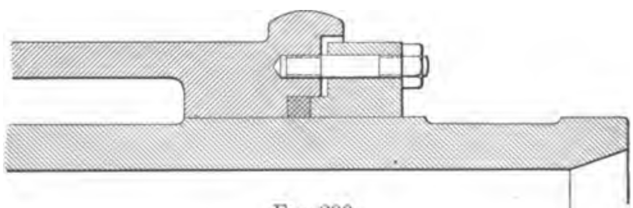


Fig. 209.

where d_1 = diameter of bolt at root of thread,
d = inside diameter of cylinder where head is attached,
p = explosion pressure, pounds per square inch,
n = number of studs,
K_t = allowable fibre stress, from 4000 to 5000 pounds per square inch.

The number of bolts may be about

$$n = 0.25\,d + 4, \qquad (3)$$

although many manufacturers used a greater number

$$n = 0.5\,d + 4, \tag{4}$$

where d is the diameter of the cylinder in inches. In order to

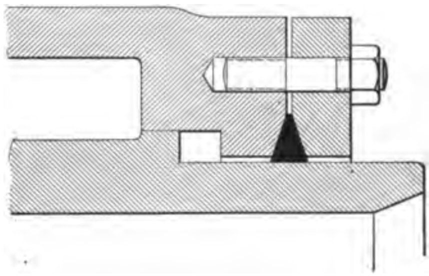


FIG. 210.

prevent the packing from being blown out the maximum distance between studs measured on the pitch circle should be 7 inches.

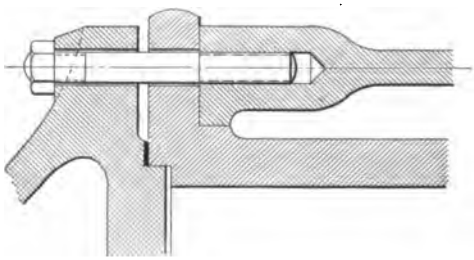


FIG. 211.

The joint between the cylinder cover and cylinder should be made tight by means of packing placed between the cover and

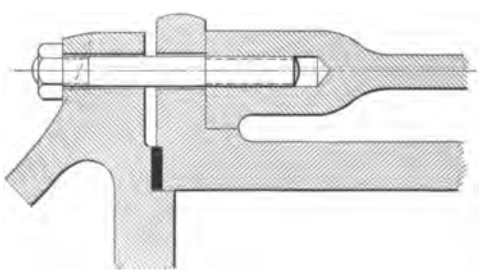


FIG. 212.

cylinder. This packing should be of non-combustible material, either asbestos or soft copper. When asbestos is used it is usually in sheet form while copper may be used in sheet or wire form.

Figs. 211, 212 and 213 illustrate forms of joint that may be used with flat packing. The first is not good as the packing is confined only by the initial tension on the bolts. In the two other cases

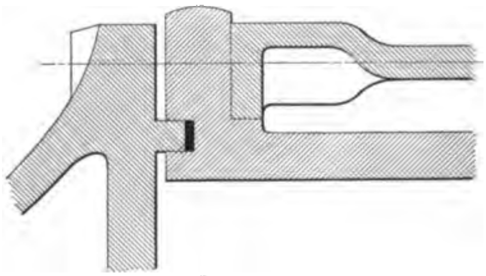


FIG. 213.

the packing is held in a recess between the cylinder and the head. Fig. 196 shows how a round copper ring may be used in place of a flat ring.

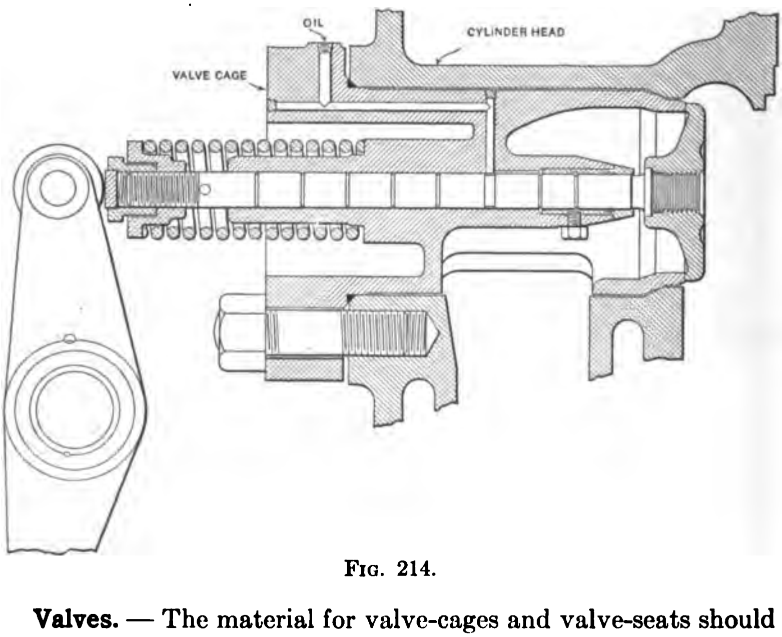


FIG. 214.

Valves. — The material for valve-cages and valve-seats should be hard close-grained cast-iron; for valve-discs medium carbon steel or nickel steel; for small automatic valves soft steel may be used. Large valves, especially those water-cooled, are made with cast-iron discs and stems of medium carbon steel.

The **allowable stress** in the valve discs should be very low on account of the high temperatures to which valves are subjected. Thus the discs will be heavy enough to guard against warping and also to allow of regrinding.

The valves of working cylinders of gas-engines today are of the poppet type. The inlet-valves may be automatic, but the exhaust-valves should be mechanically operated. For all important valves in regular and continuous operation the vertical type should be used. Horizontal valves are much more liable to leakage, sticking and other interruptions of regular service and are especially unreliable when automatic. If the use of the horizontal type is unavoidable the disc of the valve should be made light, the stem guide long and the lift as small as possible. And generally flat seats are to be preferred in horizontal valves rather than the usual conical seat. In Fig. 214 is shown a horizontal valve used as the inlet-valve of a horizontal Diesel engine. Fig. 215 illustrates a typical water-cooled exhaust-valve used on a large horizontal gas-engine. The valve-stem is made up of two concentric tubes. Water enters the bottom end of the inner tube and rises up to the hollow valve disc. Here it circulates in the disc and leaves by means of the annular space surrounding the inner tube. The water connections at the end of the valve-stem must be flexible; they are usually made with heavy rubber hose.

Design of Valves. — Let V be the volume of gas to be passed through the valve per second, v the velocity of the gas in feet per second, h the lift of the valve and d the diameter of the passage, both in feet. Then the free cross-section required through the valve in square feet will be

$$a = \pi\, dh = \frac{V}{v} \text{ square feet.} \tag{1}$$

If the lift of the valve is $h = \frac{d}{4}$, we may write

$$V = 0.7854\, d^2 v, \tag{2}$$

and from this the required diameter of the passage will be

$$d = \sqrt{\frac{V}{0.7854\, v}}. \tag{3}$$

The velocity of gas through the valve should be based on the volume of the piston displacement. This volume per second is

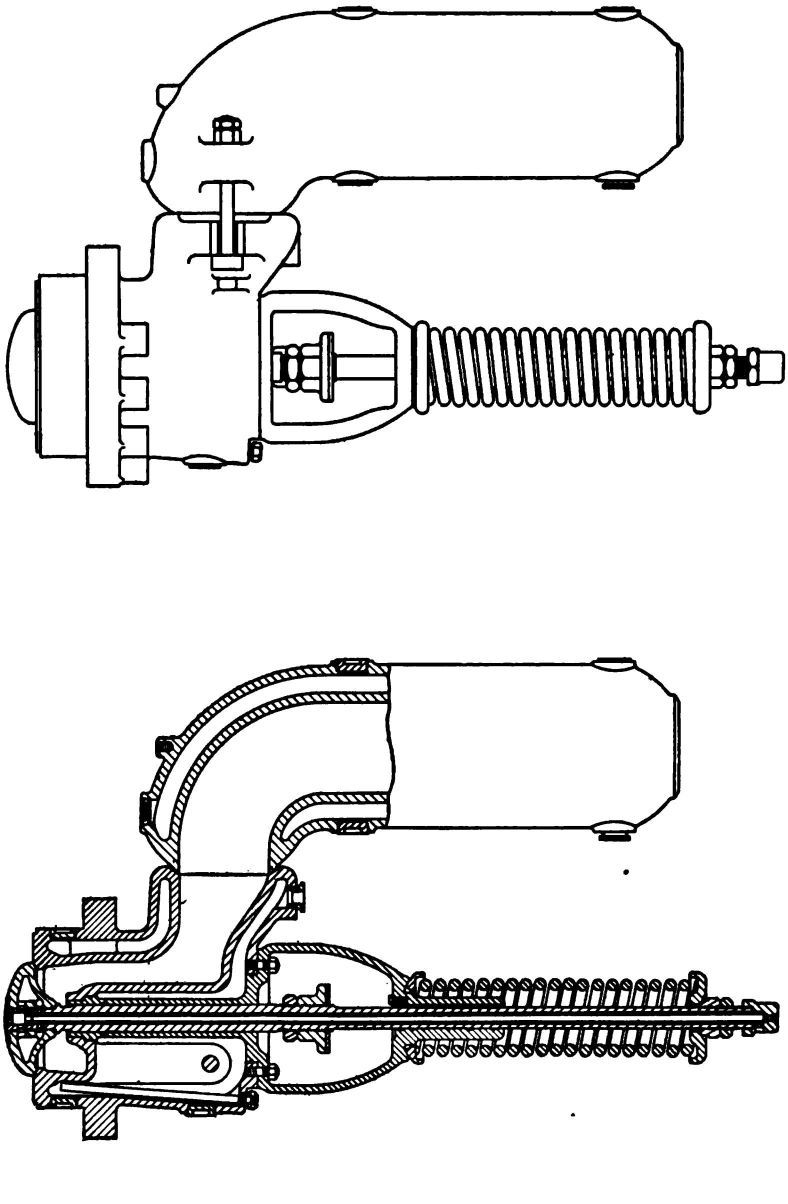


FIG. 215. — Water-cooled Exhaust-valve.

$\frac{2\,ALN}{60}$, where A is the area of the piston in square feet, L the length of the stroke in feet and N the number of revolutions per minute. Then

$$V = \frac{2\,ALN}{60}, \tag{4}$$

and

$$0.7854\,d^2v = \frac{2\,ALN}{60}. \tag{5}$$

Hence

$$d = \sqrt{\frac{2\,ALN}{60 \times 0.7854\,v}}. \tag{6}$$

The mean value of v may be assumed to be 75 feet for small engines and 100 to 120 feet for large engines. These values will answer for both inlet- and exhaust-valves.

Since the velocity of the piston is not constant the velocity of the gas through the valve will not be constant. Therefore, it is not necessary to have the valve open wide when the piston is moving at a low velocity. In Fig. 216 is shown by the heavy line

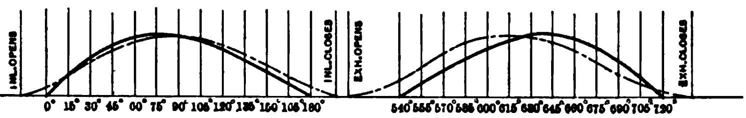


FIG. 216. — Piston Velocity and Valve-lift Curves.

a curve of piston velocity for a gas-engine, the abscissæ representing crank angles and the ordinates piston velocities. Drawn on the same base is a curve representing the lift of the valves. The inlet-valve usually opens a little before the beginning of the suction stroke and closes after the beginning of the compression stroke. This is to take advantage of any inertia that might exist in the column of gas entering the cylinder. The exhaust-valve opens when the crank is from 25 to 40 degrees before the end of the expansion stroke and remains open until after the beginning of the suction stroke. Thus the inlet- and exhaust-valves are open together for a short time and the inertia of the exhaust-gas leaving the cylinder creates a partial vacuum which assists in starting the fresh mixture into the cylinder.

The maximum lift of the valve $\frac{d}{4}$ can be utilized only in the smaller moderate-speed engines. In large engines questions of

valve design and in high-speed engines the proper instant of closing the inlet-valve make it necessary to keep the maximum lift much below $\frac{d}{4}$. Automobile engines with automatic inlet-valves usually show $h = \frac{d}{10}$ to $\frac{d}{15}$.

If the valve seat is conical instead of flat the actual lift of the valve must be increased in order to make the effective lift equal to h in equation (1). If α is the angle which the seat makes with the vertical the relation between the effective lift and the actual lift will be $\sin \alpha$ or

$$\frac{h'}{h} = \sin \alpha, \tag{7}$$

where h = actual and h' = effective lift.

The dimensions of valve-cages and guides depend on general conditions of the particular design. To make gas-tight joints between the cage and cylinder or head to which they are fitted metallic packing rings or surface ground conical at an angle of 60 degrees should be employed. In large cages the metallic packing rings are to be preferred.

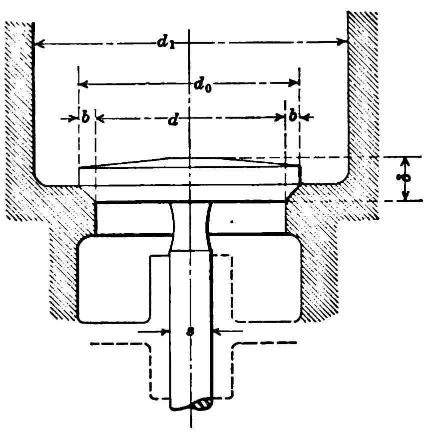


FIG. 217.

As far as the dimensions of the valve disc are concerned, besides d or d_0 only δ, Fig. 217, can be determined by computation. For mechanically operated valves with discs of medium carbon steel we may make

$$\delta = \sqrt{\frac{p_{max}\,(0.5\,d_0)^2}{6500}}\ \text{inches}. \tag{8}$$

This comes from the flat-plate formula for a circular plate supported at the edge and uniformly loaded with a pressure p_{max}, the explosion pressure. The original formula is

$$K = P\left(\frac{r}{t}\right)^2, \tag{9}$$

where K = allowable fibre stress, p = uniform pressure in pounds per square inch, r = radius of plate in inches and t = thickness of plate in inches.

For decreasing the weight of automatic valves K may be taken as high as 11,000 pounds per square inch. In discs which have a diameter greater than 4 inches, the thickness may be decreased to $\frac{2}{3}\,\delta$ at the edges.

The width b of the valve seat may be made approximately

$$b = 0.5\,(d_0 - d) = 0.01\,d + \tfrac{3}{16}\text{ inch.} \tag{10}$$

The angle of the valve seat is in most cases about 45 degrees. The diameter of the valve stem may be found from the formula

$$s = \tfrac{1}{8}\,d + \tfrac{1}{4}\text{ inch to }\tfrac{1}{8}\,d + \tfrac{7}{16}\text{ inch.} \tag{11}$$

The stem of the exhaust-valve may be made a little greater in diameter in order to decrease the wear and facilitate the conduction of heat to the cooling water. The length of the guide depends upon the design of the valve-cage or housing, and does not depend directly on the size of the valve. If short actuating levers are used on the valves longer guides are necessary than if the reverse were true.

The diameter d_1 of the valve housing must be made large enough so there will be no restriction of the passage around the valve disc. Thus

$$0.7854\,(d_1^2 - d_0^2) > 0.7854\,d^2. \tag{12}$$

This is satisfied so long as

$$d_1 \geqq 1.6\,d. \tag{13}$$

It is generally the case, however, that the form of the combustion chamber furnishes more space than is required by equation (13).

Valve Springs. — The valve-springs are generally cylindrical, helical steel springs. The requisite tension on these springs depends mainly on the vacuum occurring in the cylinder. For mechanically-operated valves the effect of this vacuum may be taken equal to a load of from 6 to 8 pounds per square inch of the cross-section of the valve. The springs should have tension enough to prevent the valves opening when the suction is as given above. Thus there will be no gas or air drawn into the cylinder at inopportune times. If the valve-springs should also be employed to bring part of the valve mechanism back into place, the inertia of these parts calls for a corresponding increase in the spring tension. This method of actuating part of the valve-

gear by means of the valve-springs is no longer considered good practice.

In designing the spring care must be taken to avoid overloading the spring when the valve is open. If, with the valve closed, the total load on the spring is W pounds, the load with the valve open will be greater than W because the spring is compressed an amount equal to the valve-lift. The amount that W is increased when the valve is opened depends on the total length of the spring, hence on the number of coils. With the valve wide open there should be from 0.05 to 0.10 inch play between the individual coils.

In discussing the actual design of the spring the following notation will be used:

S = scale of spring = load in pounds required to deflect spring 1 inch,
E = expansion or compression per coil in inches,
d = mean diameter of coil in inches,
W = load in pounds,
δ = diameter of wire or side of square in inches,
D = diameter of wire or side of square in sixteenths of an inch,
G = modulus of elasticity in torsion,
N = number of coils,
K = fibre stress in torsion caused by load W,
C = constant for deflection,
Y = constant for strength.

The relation between load and stress on a helical spring is given by the relation

$$W = \frac{K\pi\delta^3}{\frac{16\,d}{2}} = \frac{\delta^3 \times 2\,K\pi}{d \times 16} = \frac{D^3Y}{d}. \qquad (1)$$

The relation between expansion or compression per coil and the load on the spring is given by the equation

$$E = \frac{64\,W\left(\frac{d}{2}\right)^3}{G\delta^4}. \qquad (2)$$

Then $$W = \frac{E\delta^4}{d^3} \times \frac{8\,G}{64} = \frac{E\,D^4C}{d^3} \qquad (3)$$

and $$E = \frac{d^3W}{D^4C}. \qquad (4)$$

From (1) and (3)

$$d = \frac{D^3Y}{W} = \sqrt[3]{\frac{D^4CE}{W}}. \tag{5}$$

Then

$$D = \sqrt[3]{\frac{Wd}{Y}} \tag{6}$$

and

$$Y = \frac{Wd}{D^3}. \tag{7}$$

The scale of the spring is the load divided by the total deflection in inches. Then

$$S = \frac{W}{EN}. \tag{8}$$

From (4)

$$W = \frac{E\,D^4C}{d^3}.$$

Hence

$$S = \frac{E\,D^4C}{d^3EN} = \frac{D^4C}{d^3N} \tag{9}$$

and

$$N = \frac{D^4C}{Sd^3}. \tag{10}$$

In order to use the equations given above without making tiresome calculations for each spring, values of Y, C, D^3Y and D^4C are given in Tables XVIII and XIX for round and square wire of various sizes.

TABLE XVIII

		Values of Y.		Values of C.
Size of wire.	Up to $\frac{3}{8}$ inch.	$\frac{7}{16}$ to $\frac{3}{4}$ inch.	$\frac{13}{16}$ to $1\frac{1}{4}$ inches.	Up to $1\frac{1}{4}$ inches.
Stress allowed	60,000	50,000	40,000	....
Round, Steel	5.75	4.80	3.83	22
Round, Brass	2.50	....	....	11
Square, Steel	6.90	5.75	4.60	30
Square, Brass	3.00	....	....	15

TABLE XIX

Valve Spring Constants

Diameter or Side of Square No. A. W. G.	Decimals	D	D^3	D^4	D^3Y				D^4C			
					Round.		Square.		Round.		Square.	
					Steel.	Brass.	Steel.	Brass.	Steel.	Brass.	Steel.	Brass.
1¼	1.25	20	8000	160,000	30,640	....	36,800	....	3,520,000	....	4,800,000	....
1 3/16	1.188	19	6859	130,300	26,270	....	31,550	....	2,867,000	....	3,909,000	....
1⅛	1.125	18	5832	104,980	22,340	....	26,830	....	2,310,000	....	3,150,000	....
1 1/16	1.063	17	4913	83,520	18,820	....	22,600	....	1,838,000	....	2,506,000	....
1	1.	16	4096	65,540	15,690	....	18,840	....	1,442,000	....	1,966,000	....
15/16	0.9375	15	3375	50,630	12,930	....	15,530	....	1,114,000	....	1,519,000	....
7/8	0.875	14	2744	38,420	10,510	....	12,620	....	845,200	....	1,152,000	....
13/16	0.8125	13	2197	28,560	8,415	....	10,110	....	628,300	....	856,800	....
3/4	0.75	12	1728	20,740	8,294	....	9,936	....	456,200	....	622,100	....
11/16	0.6875	11	1331	14,640	6,389	....	7,653	....	322,100	....	439,200	....
5/8	0.625	10	1000	10,000	4,800	....	5,751	....	220,000	....	300,000	....
9/16	0.5625	9	729	6,561	3,500	....	4,192	....	144,300	....	196,800	....
1/2	0.5	8	512	4,096	2,458	....	2,944	....	90,110	....	122,900	....
7/16	0.4375	7	343	2,401	1,646	....	1,972	....	52,820	....	72,030	....
3/8	0.375	6	216	1,296	1,242	540	1,490	648	28,510	14,260	38,880	19,440
No.00	0.3648	5.837	198.9	1,161	1,144	497.2	1,372	596.6	25,540	12,770	34,820	17,410
11/32	0.3438	5.5	166.4	915.1	956.7	416.0	1,148	499.1	20,130	10,060	27,450	13,730
No. 0	0.3249	5.198	140.5	730.0	807.6	351.1	969.1	421.4	16,060	8,030	21,900	10,950
5/16	0.3125	5.0	125.0	625.0	718.8	312.5	862.5	375.0	13,750	6,875	18,750	9,375
No. 1	0.2893	4.629	99.19	459.1	570.3	248.0	684.4	297.6	10,110	5,050	13,770	6,887
9/32	0.2813	4.5	91.13	410.1	524.0	227.8	628.8	273.4	9,020	4,511	12,300	6,150
No. 2	0.2576	4.122	70.04	288.7	402.8	175.1	483.2	210.1	6,350	3,176	8,660	4,330
1/4	0.25	4.0	64.0	256.0	368.0	160.0	441.6	192.0	5,632	2,816	7,680	3,840
No. 3	0.2294	3.67	49.43	181.4	284.2	123.6	341.1	148.3	3,990	1,996	5,442	2,720
7/32	0.2188	3.5	42.88	150.1	246.5	107.2	295.8	128.61	3,300	1,651	4,502	2,250
No. 4	0.2043	3.269	34.93	114.2	200.9	87.3	241.0	104.8	2,512	1,256	3,426	1,713
3/16	0.1875	3.0	27.0	81.0	155.3	67.5	186.3	81.0	1,782	891	2,430	1,215
No. 5	0.1819	2.91	24.64	71.71	141.7	61.6	170.0	73.93	1,578	788.8	2151.3	1,076
No. 6	0.162	2.592	17.41	45.14	100.1	43.5	120.2	52.24	993.1	496.6	1354	677.1
5/32	0.1563	2.5	15.63	39.06	89.84	39.1	107.8	46.88	859.4	429.7	1172	586.0
No. 7	0.1443	2.309	12.31	28.43	70.78	30.8	84.94	36.93	625.4	312.7	852.8	426.4
No. 8	0.1285	2.056	8.691	17.87	49.97	21.7	59.97	26.07	393.1	196.6	536.1	268.0
1/8	0.125	2.0	8.0	16.0	46.0	20.0	55.2	24.0	352.0	176.0	480.0	240.0
No. 9	0.1144	1.83	6.129	11.22	35.24	15.32	42.29	18.39	246.7	123.4	336.5	168.2
No.10	0.1019	1.63	4.331	7.059	24.9	10.83	29.88	12.99	155.3	77.65	211.8	105.9
3/32	0.0938	1.5	3.375	5.063	19.41	8.437	23.29	10.13	111.4	55.69	151.9	75.95
No.11	0.0907	1.451	3.055	4.433	17.57	7.637	21.08	9.165	97.52	48.76	133.0	66.49
No.12	0.0808	1.293	2.162	2.795	12.43	5.404	14.92	6.485	61.50	30.75	83.9	41.95
No.13	0.072	1.152	1.529	1.76	8.79	3.822	10.55	4.586	38.72	19.36	52.8	26.4
No.14	0.641	1.026	1.108	1.108	6.21	2.7	7.452	3.24	24.38	12.19	33.24	16.62
1/16	0.0625	1.0	1.0	1.0	5.75	2.5	6.9	3.0	22.0	11.0	30.0	15.0

Example. — Design a spring for an inlet-valve which is to open downward. Diameter of valve 6 inches, weight of valve and stem 27 pounds, or 1.00 pound per square inch of valve.

The total load for which the spring is to be designed will be

$$W = \frac{\pi \times 36\ (7 + 1) \times 1.33}{4} = 292 \text{ pounds.}$$

The load on the spring with valve closed will be

$$\frac{292}{1.33} = 220 \text{ pounds.}$$

Thus the increase in load on the spring due to opening the valve is 33 per cent. If we assume the lift of the valve to be one-fifth the diameter or 1.2 inches the scale of the spring will be

$$\frac{292 - 220}{1.2} = 60 \text{ pounds.}$$

The diameter of the stem for this valve would be

$$\tfrac{6}{8} + \tfrac{3}{8} = 1\tfrac{1}{8} \text{ inches.}$$

The mean diameter of the spring d may be made 3 $\times$ the diameter of the stem or in this case $d = 3\frac{3}{8}$ inches. We may use equation (6) to find the diameter of wire necessary. Assuming the wire for the spring to be round steel with a stress of 60,000 pounds per square inch we find the value of Y to be 5.75. Then

$$D = \sqrt[3]{\frac{292 \times 3.375}{5.75}} = 5.56 \text{ sixteenths.}$$

The nearest wire would be $\frac{11}{32}$ inch diameter. Using that value we find $D^4C = 20{,}130$. Then the number of coils will be

$$N = \frac{20{,}130}{60 \times \overline{3.375}^3} = 8.75 \text{ or 9 free coils.}$$

Since the diameter of wire used, $\frac{11}{32}$ inch, does not quite agree with the calculated diameter, 5.56 sixteenths, we will use equation (7) to find the allowable load.

$$W = \frac{D^3Y}{d} = \frac{956.7}{3.375} = 284 \text{ pounds.}$$

This load is only a trifle less than the calculated load so it will not be necessary to recalculate the spring.

Allowing for a "dead" coil at each end of the spring the total number of coils will be eleven. The length of the spring with the valve open will be $11\left(\frac{11}{32} + \frac{3}{32}\right) = 4.8$ inches, allowing $\frac{3}{32}$ inch between coils when compressed. The compression of the spring due to full load will be

$$EN = \frac{W}{S} = \frac{292}{60} = 4.87 \text{ inches.}$$

The free length of the spring will then be

$$4.87 + 4.8 = 9.67 \text{ inches.}$$

The specifications of the spring will then be:

Diameter of wire........................	$\frac{11}{32}$ inch
Outside diameter of coils................	$3\frac{23}{32}$ inches
Number of free coils.....................	9
Length of spring extended...............	$9\frac{11}{16}$ inches

Design of Cams. — Exhaust- and admission-cams for a large producer gas-engine are illustrated in Fig. 218. The inlet-valve begins to open 9 degrees before head-end center on the exhaust-stroke and remains open until the crank has gone 10 degrees past crank-end dead center on the compression-stroke. The exhaust-

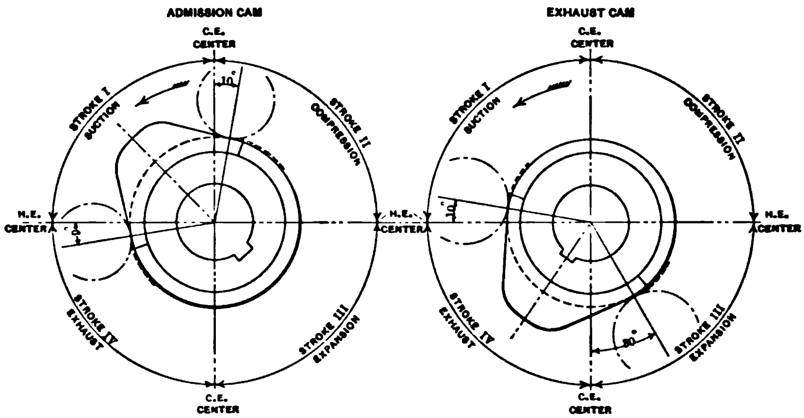


Fig. 218. — Cams for a Large Produce Gas-engine.

valve opens 30 degrees before center on the expansion stroke and remains open until the crank has gone 10 degrees past the head-end center on the suction stroke. Thus the inlet- and exhaust-valves are open at the same time for 19 degrees of the crank travel.

The cams should be made with hardened surfaces. Sometimes they are made of chilled cast-iron and sometimes of case-hardened machine steel. The rollers are usually made of hardened machine steel. The curve of the cam should be made and the roller set so there will be no jar when the cam and roller engage. There should be a slight clearance between the cam and roller when the valve is closed so that if any particles of grit or dirt fall on the cam it will pass under the roller without opening the valve at inopportune times.

Valve-gears. — To thoroughly cover the design and layout of valve-gears, as the valve operating mechanism is called, would require much more space than is available in this book. There are so many successful types of gears that it would not do to pay attention to only one particular type. Hence a short description of several types is not out of place here.

In Fig. 92 is shown a valve-gear with the inlet- and exhaust-valves in the cylinder head. The cam-shaft is above the level of the cylinder head and is driven by a vertical shaft through bevel gears. The operating arms are short, hence the valve-stems have long guides. Fig. 93 shows a different arrangement of valves in a vertical multicylinder engine. The exhaust-valve opens upward and is operated by a long push rod from the cam-shaft which is located in the base of the engine. The inlet-valve is in the head and opens downward. This valve is also operated by a long push rod through a short lever placed on top of the cylinder. Fig. 84 illustrates a valve-gear for a single-acting horizontal engine of medium size. The exhaust-valve is on the bottom of the combustion chamber and opens upward, while the inlet-valve is directly over the exhaust-valve and opens downward. Both valves are operated through levers from cams on the cam-shaft, which is off at one side and below the center-line of the engine. Each valve has its own cam. The exhaust-valve lever is a simple lever pivoted near the middle; the inlet-valve lever is also a simple lever, placed up above the level of the top of the cylinder and is horizontal. This lever is actuated by a push rod worked from a bell-crank, one arm of which carries the cam-roller. The push rod carries a knife-edge at its lower end and the position of this knife-edge along the horizontal arm of the bell-crank is determined by the governor. At full load the knife-edge is near the outer end of the bell-crank arm, thus giving the inlet-valve *A* full opening. As the load on the engine decreases the knife-edge is drawn in toward the fulcrum of the bell-crank and the opening of the valve is diminished in proportion to the load.

Fig. 128 illustrates the Snow gas-engine valve-gear. In this engine the valves are placed in a combustion chamber which is off at one side of the engine center-line. The exhaust-valve is below and opens upward; the inlet-valve is directly above the exhaust-valve and opens downward. Both valves are driven from the same cam. The cam-shaft is off at one side and on a

level with the center-line of the cylinder. The valve levers are I-shaped in section and are made of steel castings. The levers are pivoted so the lift of the valves is less than the "throw" of the cam, the ratio being about 1 to 1½ in both cases.

Fig. 120 illustrates the valve-gear of the Cooper engine. Both valves are driven from the same eccentric, the proper "lift curve" of the valves being secured by rolling levers. This method of operating the valves of large horizontal engines is becoming very popular as it presents some marked advantages over the method of operating by means of cams.

Cam-shaft and Driving Gears. — In small horizontal, single-acting engines the cam-shaft parallel to the cylinder is frequently dispensed with as the valves are arranged so they may be driven by long push rods actuated by cams on short "stub-shafts" which are geared to the main-shaft through spur gears. Cam-shafts are used on all large engines and many small ones but their design is by means of empirical formulæ rather than by rational methods. The prime requisite for a cam-shaft is rigidity and if it be designed for strength, using the loads which actually come on the shaft, it will not be rigid enough for good operation.

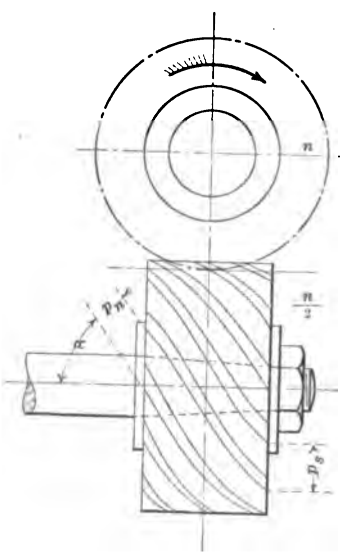


FIG. 219.

The cam-shaft is usually operated on small engines by means of cylindrical screw or spiral gears, the driver on the crank-shaft being of steel and the driven gear on the cam-shaft of bronze or cast-iron. On account of comparatively small surface of contact and the sliding friction always present in this type of gear, constant lubrication is a necessity. In order to simplify the construction of the bearings it is desirable to have the same diameter of pitch circle for both gears, or at least make the diameter of the driven gear as small as possible, in spite of the fact that the speed ratio is 2 : 1. For that reason a common tooth angle on the driving gear is 60 degrees and on the follower

30 degrees. In Fig. 219 is shown a sketch of screw gears such as are used for cam-shaft driving. The lateral pitch p_s, if p_n is the normal pitch, will then be

$$p_s = \frac{\pi D}{n} = \frac{pn}{\cos \alpha}. \tag{1}$$

For the driving gear, when $\alpha = 60$,

$$p_s = \frac{p_n}{0.5} = 2\,p_n. \tag{2}$$

For the driven gear $\alpha = 30$ and

$$p_s = \frac{p_n}{0.866} = 1.155\,p_n. \tag{3}$$

The pressure on the teeth of the driven gear causes a thrust on the cam-shaft which must be provided for at one of the bearings, preferably the one nearest the gear. The other bearing should be arranged so the relative lateral motion of the cam-shaft and cylinder, due to heating up of the latter, will not be restricted. The smaller the gear angle on the driven gear the less will be the thrust on the cam-shaft. The width of the teeth or face of the gears is usually from 2 to 2½ p_n. Table XX gives average dimensions of valve-gear parts for small engines.

TABLE XX

Average Dimensions of Valve-gear Parts

B.H.P. of engine.	2	5	10	15	20	30	40	50	60	75	100
p_n	0.65	0.68	0.68	0.71	0.71	0.74	0.74	0.80	0.80	0.87	0.93
d_s	1¼	1⅜	1⅜	1⅝	1⅝	1¾	1¾	2	2	2¼	2½
d_r	1⅝	1¾	2	2¼	2⅜	2⅝	2¾	3	3¼	3½	4

p_n = normal pitch of gears,
d_s = diameter of cam-shaft,
d_r = diameter of cam-roller, all dimensions in inches.

Cam-shaft gears for large engines require more careful consideration than those for small engines. Many manufacturers do not use spiral gears but prefer to use bevel gears which are sometimes used in connection with spur gears, thus utilizing two pairs of gears to get the 2 : 1 ratio. In this way each ratio may be made a fractional number with the result that there will be a "hunting tooth" in the system, that is, the load which is applied

suddenly as the valve is opened will not always come on the same tooth but will come on all the teeth in turn. With this method the gears wear evenly and last much longer than if a straight 2 : 1 ratio in one pair of gears is used.

Example.—If a cam-shaft is driven from a crank-shaft through a pair of spur gears with 24 teeth on the driving and 25 teeth on the driven gear and a pair of bevel gears with 25 teeth on the driver and 48 teeth on the follower, the total ratio will be

$$\frac{24 \times 25}{25 \times 48} = 2$$

and the wear on the gears will be distributed over all the teeth.

EXERCISES

1. Find the stress in a gas-engine cylinder 14 ins. diameter when the explosion pressure is 400 lbs. per sq. in. Thickness of cylinder $1\frac{1}{4}$ ins. (*a*) Due to radial pressure alone. (*b*) Due to radial and longitudinal pressure when Poisson's ratio is 0.25.

2. Find the thickness of cylinder by two methods when the diameter is 16 ins. and pressure is 390 lbs. per sq. in. Find thickness of flange and jacket wall to correspond.

3. Referring to Fig. 195: $D = 18$ ins., $T = 10{,}000$ lbs., $l = 24$ ins., $c = 14$ ins., $c_1 = 18$ ins., $b = 26$ ins., $b_1 = 30$ ins., $t = 1\frac{1}{2}$ ins. Find stress in cylinder due to bending.

4. Referring to Fig. 200: $P = 35{,}000$ lbs., $b = 3$ ins., $h = 30$ ins., $l = 4$ ins., $f = 2$ ins. Find stress in frame.

5. Referring to Fig. 201: $P = 40{,}000$ lbs., $b = 3\frac{1}{2}$ ins., $h = 32$ ins., $c = 24$ ins., $g = 22$ ins., $k = 2$ ins., $f = 2$ ins. Find stress in frame.

6. Compute the number and diameter of studs for a gas-engine cylinder 24-in. diameter, pressure 375 lbs. per sq. in.

7. Compute dimensions of valve and stem for a 20×28-in. gas-engine running 160 r.p.m. Pressure in cylinder 400 lbs. per sq. in. (*a*) With flat seat. (*b*) With conical seat.

8. Compute spring for an 8-in. valve, valve and stem to weigh 45 lbs. Allow 8 lbs. per sq. in. of valve in addition to weight for load on spring. Spring to be round steel with a working stress of 50,000 lbs. per sq. in. Load on spring to increase 25 per cent when valve opens $1\frac{1}{2}$ ins.

CHAPTER XXIII

THE CRANK-SHAFT

Side-cranks — Center-cranks — Multiple-cranks

General. — The **crank-shaft** is one of the most vital parts of the engine and hence requires careful consideration in design. No matter what the type of engine, the shaft may be considered as a beam with two or more supports. In addition to the **bending** stress there is always a **shearing** stress present at some time during the stroke caused by the twisting moment on the shaft. Both the bending and the twisting moments are due to the pressure in the cylinder. In addition there are bending moments due to the weight of the flywheel, the weight of the rotor when an engine is direct-connected to an electric generator and the pull of the belts when the engine is belted to a machine of any kind.

It is usually not difficult to determine the magnitudes of the various loads on a crank-shaft but it is very often difficult to determine the distance between supports and the location of the loads in reference to the supports. It is usual to consider the shaft **supported** at the centers of the bearings. The distance between supports then depends on the length of the bearings and the length of the bearings depends on the diameter of the shaft, which necessarily depends on the distance between the supports. This makes it necessary to make an assumption of the lengths of the different bearings before the moments can be calculated. When the diameter is found the lengths of bearings must be recalculated and if the new dimensions are far different from the assumed ones, the diameter of the shaft must be recalculated. New assumptions must be made and new diameters calculated until the assumption agrees closely with the calculated diameter.

The method of calculation necessarily varies with the type of engine. The center-crank type of shaft requires calculations that are entirely different from those employed with a side-crank

type. Likewise a single-crank shaft requires calculations different from those used with a two-crank or multiple-crank shaft. On account of the numberless combinations of cranks and bearings that we might make it is obviously impossible to cover them all in a general book on gas-engine design. In this book we will cover two or three common types, a side-crank, a center-crank and a multiple-crank shaft. If one is familiar with these three types it will not be difficult to design any other as the theory is the same in all.

There are several ways of finding the bending moments at various points on a beam carrying more than one load. One way is to find the net reactions at the supports due to all the loads. Another method is to find the reactions and bending moments due to the individual loads separately, then combine the bending moments. The author prefers the latter method. It is usually most convenient to find the bending moment for each load by calculation and then draw the individual bending moment diagrams and find the ordinates of the combined bending moment diagram graphically.

The **material** used in crank-shafts is usually open-hearth steel with a tensile strength of 64,000 pounds per square inch and an elongation of at least 20 per cent in 8 inches or mild crucible steel with a tensile strength of 70,000 pounds per square inch and an elongation of 18 or 20 per cent in 8 inches. Of late years nickel steel has been used very freely but it has not been entirely satisfactory as it apparently becomes brittle under repeated straining such as it is subjected to in crank-shafts. The crank-shafts of automobile and other high-speed multicylinder motors are frequently made of nickel steel with chromium added. This steel usually has a tensile strength of from 130,000 to 140,000 and an elastic limit of 110,000 pounds per square inch. The elongation in 2 inches is about 20 per cent and the reduction of area 55 per cent.

The Side-crank Shaft. — This type of shaft is used largely in America for medium-sized and large engines, but does not seem to be in favor in England nor on the continent of Europe. Its chief advantage is that it requires only two bearings no matter whether the engine is a single- or twin-cylinder unit.

The **crank, pin** and **shaft** may be forged in one solid piece or they may be made separately and forced together with hydraulic

pressure. The latter method is usually employed in large engines, the former in medium-sized engines. Fig. 220 shows a typical side-crank forged solid with the shaft. The balance weight, in this case, would have to be bolted to the crank. This method is expensive and is not used on large work. The type shown in

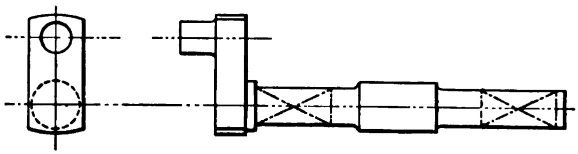


FIG. 220. — Side-crank-shaft Forged in One Piece.

Fig. 221 is used freely. The crank-arm and counterweight are usually made of cast-steel. The crank-pin and the shaft are forced into place under hydraulic pressure. Some manufacturers cast the pin integral with the crank and weight. This method requires a better grade of material in the crank and weight than would be necessary if the pin and crank were made separate.

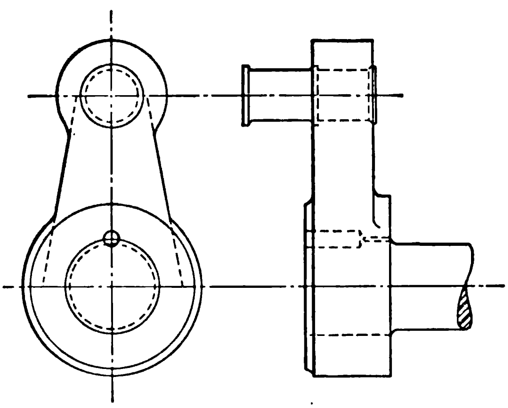


FIG. 221. — Side-crank with Crank Cast Separately.

The crank-shaft should be designed or investigated for at least two positions of the crank, one with the crank-pin on the center at the beginning of the explosion stroke, the other with the crank at the angle where the twisting moment is a maximum. The first case considered here will be with the crank on the center, the shaft to have a flywheel whose weight is W and the flywheel is to be a belt-wheel also, the belt to be horizontal. The distribu-

tion of loads will be as shown in Fig. 222. The size of the crank-pin has been discussed in Chapter XX. The thickness of the crank-arm e may be assumed to be equal to $\frac{3}{4}\,d_1 + 1$ inch, where d_1 is the diameter of pin, and the distance l equal to $2\frac{1}{4}\,d_1$. Then

$$a = \frac{l_1}{2} + e + \frac{l}{2}.$$

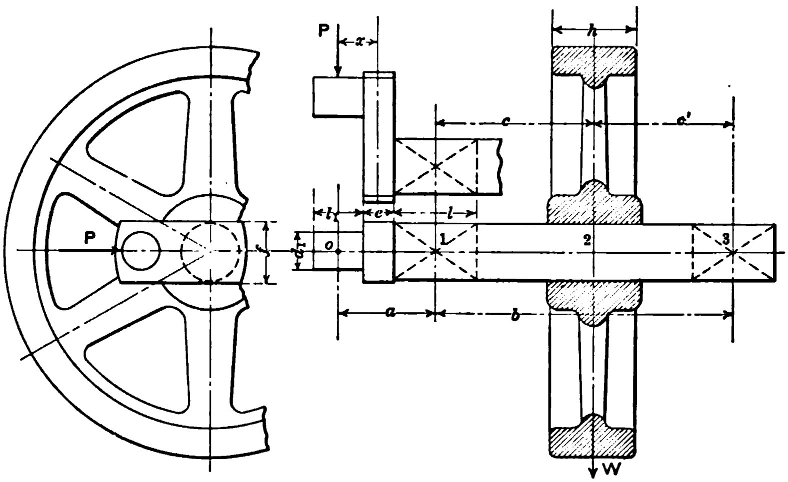


FIG. 222.

The distances c and c' will be determined by the width of the fly-wheel and space required for gears or eccentrics. The distance c may be found from

$$c = l + \frac{h}{2}.$$

This will leave a distance of $\frac{l}{2}$ inches between the edge of the fly-wheel and the edge of the bearing for gears. The distance c' may be made less than c, but in many cases is made equal to c.

The diameters at the center of bearing 1 and at the center of the flywheel at 2 should be determined. All loads will be assumed to be in pounds and all dimensions in inches. Using first the load P, the explosion pressure multiplied by the area of the piston, we find the bending moment at 1 to be

$$M_b = Pa \text{ inch-pounds,} \tag{1}$$

in the horizontal plane if the engine is horizontal. The bending moment at 2 in the horizontal plane due to the belt pull will be

$$M_{2h} = \frac{(T_1 + T_2)\,cc'}{c + c'} \text{ inch-pounds.} \tag{2}$$

The bending moment at 2 in the vertical plane will be

$$M_{2v} = \frac{Wcc'}{c + c'} \text{ inch-pounds.} \tag{3}$$

The bending moments found in (1), (2) and (3) should now be laid off on a base representing the distance $a + b$. This horizontal scale may be made anything that is convenient, usually $1\frac{1}{2}$ or 3 inches = 1 foot. The vertical scale, inch-pounds per inch, should be assumed such as to give a maximum height of diagram of about 3 inches. The bending moment diagram of the shaft shown in Fig. 222 is given in Fig. 223. As a matter of convenience the

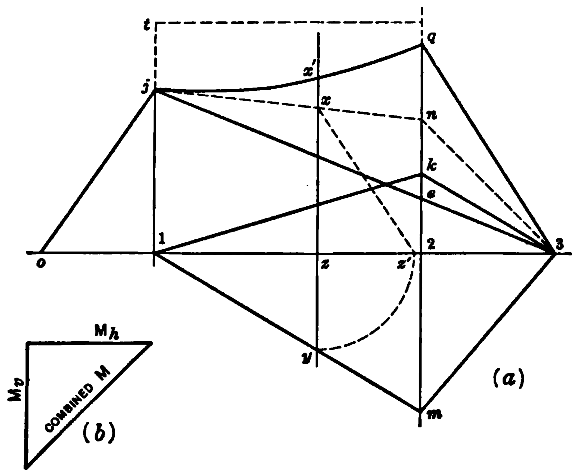


FIG. 223.

vertical moments are laid off below and the horizontal moments above the line. The diagram 1 *m* 3 is for the flywheel, 2 *m* being laid off equal to the moment given by equation (3). Diagram 1 *k* 3 is for the belt pulls, 2 *k* being equal to the moment given by equation (2) and *oj* 3 for the piston thrust, 1 *j* being equal to the moment given by equation (1). Since *oj* 3 and 1 *k* 3 are horizontal and tend to deflect the shaft in the same direction, they are added. To do this it is only necessary to lay off *kn* equal to

$2e$ and draw in the dotted diagram $jn\,3$. The ordinates of $ojn\,3$ then represent the total horizontal moments on the shaft at the various points along the axis of the shaft.

To find the diagram of total bending moments it will be necessary to combine $1\,m\,3$ with $ojn\,3$. The most convenient way to do this is to lay off the vertical moment for a given point on the shaft and at right angles to it lay off the corresponding maximum horizontal moment. The closing side of the triangle will then give the magnitude and direction of the combined moment. This is shown at b, Fig. 223. Since the direction of the combined moment is of no importance in this particular problem, the combination of the vertical and horizontal moments may be performed on the diagram at a, Fig. 223. The method is this: at any point along the base draw an ordinate such as xy, cutting the base at z. With compasses drawn in the arc yz' as shown, xz' is the combined moment and may be laid off on the ordinate xy from z, giving us the point x'. Performing this operation for several points we get the diagram $ojq\,3$, **the ordinates of which represent the total combined bending moments for all points along the shaft.**

In the case illustrated above the maximum moment is $2\,q$ under the flywheel. Using this moment we find the diameter of the shaft to be

$$d = \sqrt[3]{\frac{32\,M_{2q}}{\pi K_t}} \text{ inches.} \tag{4}$$

The condition of loading on this shaft is very severe; the stress on the extreme fibre varies from a maximum tension to a maximum compression and back to a maximum tension with each revolution of the shaft. This rapid reversal of stress necessitates a low fibre stress of 6000 pounds per square inch. The deflection of the shaft also influences the diameter as will be shown later in this chapter. A higher fibre stress would result in a shaft in which the deflection would be extreme.

Stress in the Crank-arm. — The bending moment on the crank-arm is Px, where $x = \frac{l_1}{2} + \frac{e}{2}$. The resisting moment of the arm is $\frac{K_t f e^2}{6}$. The stress will then be

$$K_t = \frac{6\,Px}{fe^2} \text{ pounds per square inch.} \tag{5}$$

The value of f may be assumed to be from $\frac{1}{2}$ to 1 inch greater than the shaft diameter in the bearing. In addition there is a direct compression stress in the crank-arm which is

$$K_c = \frac{P}{fe} \text{ pounds per square inch.} \qquad (6)$$

The total stress on the fibres of the side of the crank-arm near the crank-pin will be

$$K_{\text{total}} = K_t + K_c. \qquad (7)$$

This total stress should not exceed 10,000 pounds per square inch.

Crank in Position of Maximum Twisting Moment. — After the calculations are made with the crank on the dead center as in the preceding article, the crank angle θ where the maximum twisting moment occurs should be found. This may be done graphically by taking the pressure from the indicator card for different crank angles and finding P_t, Fig. 224. The crank angle which gives the largest value of P_t should be used for θ, Fig. 224.

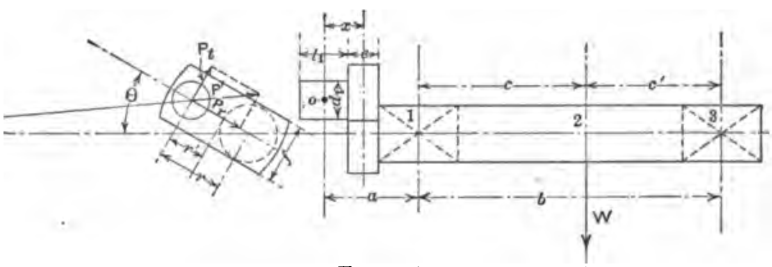


FIG. 224.

If it were not for the fact that the inertia of the reciprocating parts should be neglected in finding P_t maximum we might take the value directly from the rotative effort diagram. During the time in which the engine is starting the inertia forces are small enough to be neglected but the pressure is a maximum during that time. Hence we use the pressure from the card. The angle θ at which P_t is a maximum will probably be between 30 and 40 degrees. After that angle is found a sketch similar to that in Fig. 224 should be drawn and the forces P_r and P_t, radial and tangential components of P' respectively, should be put on, bearing in mind that P' is taken from the indicator card at the piston position corresponding to the crank-angle θ, and that $P' = \frac{p\pi D^2}{4} \sec \alpha$ where p is the pressure per square inch on the piston at crank-angle θ,

d is the diameter of the piston in inches, and α is the angle which the connecting-rod makes with the center-line of the engine.

When a value of $5\frac{1}{2}$ or 6 for the ratio of the length of the connecting-rod to the crank is used, as in most stationary engines, the force P' will be so nearly horizontal that we may call it so without appreciable error. Then the bending moments may be found and combined in precisely the same manner as in Fig. 223, using P' as the load. In addition there will be a twisting moment $P_t r$ on the shaft extending from bearing 1 to the center of the flywheel as shown by the dotted horizontal line in Fig. 223. This constant twisting moment should be combined with the maximum bending moment $2\,q$ to find the equivalent moment.

There are several ways of finding the equivalent bending moment when there are stresses at right angles to each other, as is the case when a shaft is under bending and torsion at the same time. The torsion causes shear on a plane normal to the axis of the shaft and the bending causes tension and compression on a plane parallel to the axis of the shaft. A general condition is illustrated in Fig. 225 which shows direct stresses on the planes 1–2, 4–3, 2–3 and 1–4 of an elementary cube with shear on the planes as shown at right angles to the direct stresses K_t and K_t'. There will be some plane as ab making an angle α with 2–3 on which the stress will be entirely normal. This normal stress is called the **principal stress** and is the greatest stress to which the material is subjected. It can be shown that

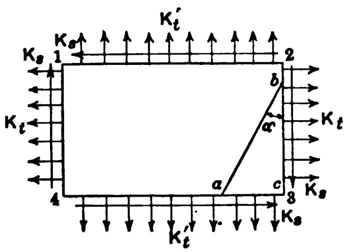


Fig. 225.

$$\tan 2\alpha = \frac{2\,K_s}{K_t - K_t'}, \tag{1}$$

from which two values of α differing by a right angle may be found. It can also be shown that

$$K_{t_{\max}} = \tfrac{1}{2}\,(K_t + K_t') \pm \sqrt{\tfrac{1}{4}\,(K_t - K_t')^2 + K_s^2}. \tag{2}$$

These two values of $K_{t_{\max}}$ are the values of the normal stress intensities on the two principal planes. The larger value (where the upper sign is taken) will be the stress intensity on such a plane as a–b, Fig. 225, and will be the same sign as K_t and K_t'.

The planes on which there are maximum shear stresses are inclined at 45 degrees to the principal planes found and the maximum intensity of shear is

$$K_{s_{max}} = \sqrt{\tfrac{1}{4}(K_t - K_t')^2 + K_s^2}. \tag{3}$$

The condition existing in the engine shaft is such that K_t' is zero. Then

$$K_{t_{max}} = \tfrac{1}{2}K_t + \sqrt{\tfrac{1}{4}K_t^2 + K_s^2}. \tag{4}$$

Since $$K_{t_{max}} = \frac{M_e}{Z},$$

$$K_t = \frac{M}{Z},$$

and $$K_s = \frac{M_t}{Z_p}.$$

We may write

$$\frac{M_e}{Z} = \frac{1}{2}\frac{M}{Z} + \sqrt{\frac{1}{4}\left(\frac{M}{Z}\right)^2 + \left(\frac{M_t}{Z_p}\right)^2}, \tag{5}$$

where M_e is the **equivalent** bending moment causing the maximum or principal normal stress, M is the bending moment causing the stress K_t, M_t is the twisting moment causing the shearing stress K_s, Z is the section modulus $\frac{I}{d_1}$ of a shaft in bending and Z_p is the section modulus of a shaft in torsion. Since $Z_p = 2\,Z$ we may write

$$\frac{M_e}{Z} = \frac{1}{2}\frac{M}{Z} + \sqrt{\frac{1}{4\,Z^2}(M^2 + M_t^2)}, \tag{6}$$

whence $$M_e = \tfrac{1}{2}M + \tfrac{1}{2}\sqrt{M^2 + M_t^2}. \tag{7}$$

Thus, referring to Fig. 223, the bending moment 2–q may be substituted for M and the twisting moment 1–t for M_t in (7) and the result M_e will be a bending moment which would cause the maximum normal stress on the principal plane a–b, Fig. 225. From this equivalent bending moment may be found the diameter of the shaft

$$d = \sqrt[3]{\frac{32\,M_e}{\pi K_t}}\ \text{inches}. \tag{8}$$

On the continent of Europe it has been usual to consider the safety of a bar subjected to bending and twisting moments to

depend on the greatest tensile strain or stretch. Then if Poisson's ratio $= \frac{1}{4}$, we have for a circular section

$$M_e = \tfrac{3}{8} M + \tfrac{5}{8} \sqrt{M^2 + M_t^2}. \tag{9}$$

Experiments by Mr. J. J. Guest on ductile materials subjected to combined tension and shear appear to show that failure begins when the shearing stress attains a definite value. While the maximum tension at which yielding occurred in the specimens used by Mr. Guest varied considerably the maximum shearing stress in the different tests was nearly constant. Using equation (3) remembering that K_t' is zero we get

$$K_{s_{\max}} = \sqrt{\tfrac{1}{4} K_t^2 + K_s^2}, \tag{10}$$

and

$$\frac{M_{T_e}}{Z_p} = \sqrt{\frac{1}{4}\left(\frac{M}{Z}\right)^2 + \left(\frac{M_t}{Z_p}\right)^2}. \tag{11}$$

But $Z_p = 2\,Z$. Then

$$\frac{M_{T_e}}{Z_p} = \frac{1}{2\,Z}\sqrt{M^2 + M_t^2}, \tag{12}$$

whence

$$M_{T_e} = \sqrt{M^2 + M_t^2}. \tag{13}$$

This result M_{T_e} is the equivalent **twisting** moment and should be used with the resisting moment in **torsion** in order to find the diameter of a shaft. Then

$$d = \sqrt[3]{\frac{16\,M_{T_e}}{\pi K_s}}\ \text{inches.} \tag{14}$$

It should be emphasized here that K_s is the allowable intensity of shear, about 4000 or 5000 pounds per square inch for material used in gas-engine shafts.

Stress in the Crank-arm. — The stresses in the crank-arm are more complicated with the shaft in position of maximum turning moment than with the crank on the center. The section where the arm joins the shaft will be the most dangerous as at that point there will be direct compression due to P_r, bending due to P_r and bending due to P_t. The direct compression will be

$$K_c = \frac{P_r}{ef}, \tag{1}$$

the bending stress due to P_t will be

$$K_b = \frac{6\,P_t r'}{ef^2} \tag{2}$$

and that due to P_r will be

$$K_b' = \frac{6\,P_r x}{e^2 f}. \tag{3}$$

The maximum stress will be at a corner on which both the bending stresses found by (2) and (3) will be compression. To these two must be added the direct compression or

$$K_{\text{total}} = K_c + K_b + K_b'. \tag{4}$$

In addition to the compressive stress there is a torsional stress due to the twisting moment $P_t x$. The polar moment of inertia of a rectangle is the sum of the moments of inertia about the two principal axes at right angles. Thus, for the crank under consideration, the polar moment would be

$$I_p = \frac{ef^3}{12} + \frac{e^3 f}{12} = \frac{ef}{12}(e^2 + f^2). \tag{5}$$

Theoretically the distance from the neutral axis to the remote fibre is

$$d_1 = \sqrt{\left(\frac{e}{2}\right)^2 + \left(\frac{f}{2}\right)^2}, \tag{6}$$

making the section modulus

$$\frac{I_p}{d_1} = \frac{\frac{ef}{12}(e^2 + f^2)}{\frac{1}{2}\sqrt{e^2 + f^2}} = \frac{ef}{6}\sqrt{e^2 + f^2}. \tag{7}$$

Actually the fibre at the corner of the section of a rectangular shaft in torsion is not under the greatest stress but the greatest stress falls on the fibre at the middle of the long side. The section modulus is given by Morley's "Strength of Materials," as

$$\frac{I}{d_1} = \frac{a^2 b}{3 + 1.8\frac{b}{a}}, \tag{8}$$

where a is the short and b the long side.

Unwin gives the section modulus as

$$\frac{I}{d_1} = \frac{2}{9}a^2 b. \tag{9}$$

Using the last value we find the torsional stress in the crank-arm to be

$$K_s = \frac{9\,P_t x}{2 f e^2}. \tag{10}$$

In order to find the maximum combined stress in the arm we may use the equation

$$K_{c_{max}} = \frac{1}{2} K + \sqrt{\frac{1}{4} K^2 + K_s^2}, \tag{11}$$

where K = the total stress found by equation (4) and K_s = shearing stress found by equation (10). This maximum stress should not exceed 10,000 pounds per square inch.

The diameter of the shaft in the bearings should be determined by the same method used for finding the diameter under the flywheel. At bearing 1 the bending moments due to flywheel and belt pulls will be zero but there will be bending and torsion due to the piston pressure. If the equivalent bending moment is less at 1 than at 2, Fig. 224, the shaft diameter is sometimes made less to correspond but the diameter under the flywheel at 2 is never made less than in the bearing at 1. Usually the bearing at 3 is made the same diameter as at 1. The reactions due to the piston pressure, belt pulls and flywheel at the bearings should be found and the length of the bearings determined, using an allowable pressure per square inch of projected area. The calculations for this work will be found in the latter part of this chapter.

After the lengths of the bearings have been determined new values of a, b, c, c', e, f and x, Fig. 222, may be computed and the shaft diameters recalculated as before. Since these second values will be more accurate than the first assumptions, the second set of diameters found will, in all probability, be the true diameters. However, the dimensions mentioned above should be calculated once more after the second diameters are found and if there is much change a third set of diameters must be found.

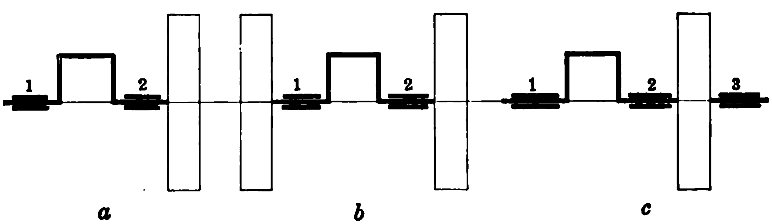


FIG. 226. — Types of Shafts used on Center-crank Engines.

The stresses found in the crank-arm will determine whether any change may be made in e and f. It is obvious that f must be greater than the diameter of the shaft at bearing 1.

The Center-crank Shaft. — There are at least three types of crank-shafts used with single- or tandem-cylinder center-crank engines. These are illustrated in Fig. 226. (*a*) and (*b*) are used only on small engines while (*c*) is used on either small or large engines. In this discussion we will confine ourselves to the type (*c*). The principles of all three are the same and an understanding of one will enable a student to calculate the other two.

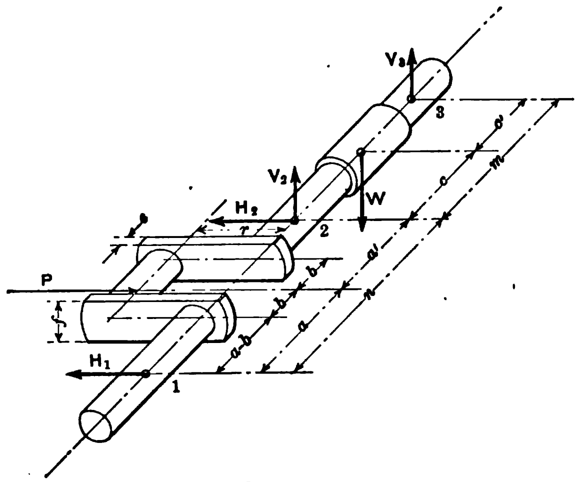


FIG. 227.

Crank on Center. — The shaft of a horizontal engine with its loads and dimensions will appear as in Fig. 227. It is the usual practice to consider that the piston pressure does not have any influence in the bending stress between bearings 2 and 3 and that the weight of the flywheel and the pull of the belt do not have any influence on the bending between bearings 1 and 2.

The Crank-pin. — The bending-moment at the center of the crank-pin is

$$M_b = H_1 a \text{ inch-pounds,} \tag{1}$$

$$H_1 = \frac{Pa'}{n} \text{ pounds.} \tag{2}$$

The shaft is usually made so that a and a' are equal. Then the bending moment at the center of the crank-pin is

$$M_b = \frac{Pn}{4} \text{ inch-pounds.} \tag{3}$$

The length n should be assumed to be 1.8 to 2.2 D, where D is the diameter of the cylinder.

The diameter of the crank-pin will be

$$d = \sqrt[3]{\frac{32\, M_b}{\pi K_t}} \text{ inches.} \tag{4}$$

The allowable stress should be 10,000 or 12,000 pounds per square inch.

Left-hand Crank-arm. — After the diameter of the crank-pin has been determined as above, the dimensions of the crank-arm may be determined. The width of the arm f may be made $1\frac{1}{8}\, d + \frac{1}{2}$ inch and the thickness $e = 0.65\, d + \frac{1}{4}$ inch. The bending moment on the arm is

$$M_b = H_1\,(a - b) \text{ inch-pounds.} \tag{1}$$

From this the stress may be found

$$K_b = \frac{6\, M_b}{e^2 f} \text{ pounds per square inch.} \tag{2}$$

The stress due to direct compression will be

$$K_c = \frac{H_1}{ef}.$$

The total stress will be

$$K_{\text{total}} = K_b + K_c.$$

This stress should not exceed 10,000 pounds per square inch.

Right-hand Crank-arm. — The bending moment on the right-hand crank-arm will be

$$M_b = H_1\,(a + b) - Pb \text{ inch-pounds,} \tag{1}$$

which is the same as that on the left-hand arm, and since the size of the arm is the same the stress will be the same. This stress is on the face of the broad side of the arm.

Shaft Under Flywheel. — The moments due to the flywheel in the vertical and the belt pull in the horizontal plane may be found as for the side-crank shaft. These two moments should be combined at right angles and the corresponding diameter of the shaft found. Thus if M_w is the bending moment due to the flywheel and M_b that due to the belt the total bending moment will be

$$M_{\text{total}} = \sqrt{M_w^2 + M_b^2} \text{ inch-pounds.} \tag{1}$$

The diameter of the shaft will be

$$d = \sqrt[3]{\frac{32\,M_{\text{total}}}{\pi K_t}} \text{ inches,} \tag{2}$$

where K_t is the allowable stress of 6000 pounds per square inch.

Crank at Angle of Maximum Twisting Moment. — Fig. 228 illustrates the center-crank shaft at a position where the twisting moment is a maximum. This position should be found as stated in the discussion of the side-crank shaft. After the total force P' is found for this position, the tangential and horizontal components P_t and P_r should be found graphically.

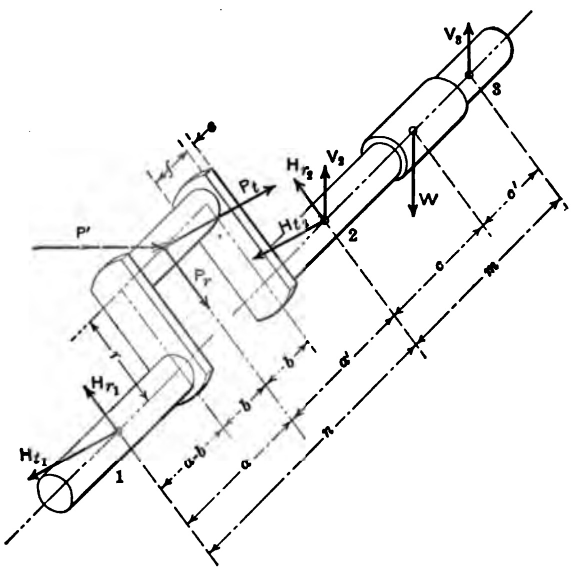


Fig. 228.

Shaft Under Flywheel. — The maximum combined bending moment $M_{b_{\max}}$ due to the weight of the flywheel and the pull of the belt should be found as before. The twisting moment will be

$$M_t = P_t r \text{ inch-pounds.} \tag{1}$$

The equivalent twisting moment will be

$$M_{te} = \sqrt{M_t^2 + M_{b_{\max}}^2} \text{ inch-pounds.} \tag{2}$$

Using this moment we find the diameter of the shaft under the flywheel to be

$$d = \sqrt[3]{\frac{16\, M_{t_e}}{\pi K_s}} \text{ inches,} \tag{3}$$

where K_s is the allowable stress in shear, about 5000 pounds per square inch.

Shaft at Juncture of Right-hand Crank-arm. — The twisting moment here is the same as that given under the flywheel. The bending moment will be

$$M_b = H_1\left(a + b + \frac{e}{2}\right) - P'\left(b + \frac{e}{2}\right) \text{inch-pounds,} \tag{1}$$

$$\text{where } H_1 = \frac{P'}{2}, \text{ the reaction at 1 due to } P'.$$

The twisting and bending moments should be combined to find the equivalent twisting moment using Guest's law as before, and this equivalent moment used to find the diameter using a stress of 5000 or 6000 pounds.

Crank-pin. — The crank-pin is subjected to bending and twisting in the position shown in Fig. 228. The bending moment is due to H_1 and is

$$M_b = H_1 a \text{ inch-pounds.} \tag{1}$$

The twisting moment is due to H_t and is

$$M_t = H_{t_1} r \text{ inch-pounds.} \tag{2}$$

The equivalent twisting moment may be found by combining (1) and (2) according to Guest's law. Then

$$M_{t_e} = \sqrt{M_b^2 + M_t^2} \text{ inch-pounds.} \tag{3}$$

Then the diameter will be

$$d = \sqrt[3]{\frac{16\, M_{t_e}}{\pi K_s}} \text{ inches,} \tag{4}$$

where K_s is the allowable shearing stress in pounds per square inch, about 5000 or 6000.

Left-hand Crank-arm. — The left-hand crank-arm is subjected to bending due to H_{r_1} and H_{t_1}, to twisting due to H_{t_1} and to direct compression due to H_{r_1}.

The bending moment due to H_{r_1} is

$$M_{b_r} = H_{r_1}(a - b) \text{ inch-pounds.} \tag{1}$$

The corresponding stress is

$$K_{b_r} = \frac{6\, M_{b_r}}{e^2 f} \text{ pounds per square inch.} \tag{2}$$

The bending moment due to H_{t_1} is

$$M_{b_t} = H_{t_1} r' \text{ inch-pounds,} \tag{3}$$

at the juncture of the crank-pin and arm, where $r' = r - \frac{1}{2}$ diameter of crank-pin.

The corresponding stress is

$$K_{b_t} = \frac{6\, M_{b_t}}{e f^2} \text{ pounds per square inch.} \tag{4}$$

The stress in direct compression is

$$K_c = \frac{H_{r_1}}{e f} \text{ pounds per square inch.} \tag{5}$$

The total compression stress on the fibres on one corner will be

$$K_{c_{\text{total}}} = K_{b_r} + K_{b_t} + K_c. \tag{6}$$

In addition to the above stresses there is a stress in torsion. The twisting moment is

$$M_t = H_{t_1} (a - b) \text{ inch-pounds.} \tag{7}$$

The corresponding stress is

$$K_s = \frac{9\, M_t}{2 f e^2} \text{ pounds per square inch.} \tag{8}$$

The total combined stress on the arm will then be

$$K_{c_{\max}} = \frac{1}{2} K + \sqrt{\frac{K^2}{4} + K_s^2}, \tag{9}$$

where K = the total stress found by equation (6) and K_s = shearing stress found by equation (8). The maximum stress as found by equation (9) should not exceed 10,000 pounds per square inch.

Right-hand Crank-arm. — The crank-arm on the right or towards the flywheel is subjected to bending stresses in two planes normal to each other and also to torsion. The bending moment due to the radial pressure is constant throughout the length of the crank-arm. It is

$$M_{b_r} = H_{r_1} (a + b) - P_r b \text{ inch-pounds.} \tag{1}$$

The stress to correspond is

$$K_{b_r} = \frac{6\, M_{b_r}}{e^2 f} \text{ pounds per square inch.} \tag{2}$$

The bending due to the tangential component is a maximum where the arm joins the shaft, hence the total stress will be a maximum at that point. The bending due to the tangential component is

$$M_{b_t} = P_t r'' + H_{t_1} \times \text{radius of shaft at 2 inch-pounds}, \qquad (3)$$

where $r'' = r - \frac{1}{2}$ diameter of shaft at 2.

The stress to correspond will be

$$K_{b_t} = \frac{6\, M_{b_t}}{ef^2} \text{ pounds per square inch.} \qquad (4)$$

The stress in direct crushing is

$$K_c = \frac{P_r}{2\, ef} \text{ pounds per square inch.} \qquad (5)$$

The twisting moment on this arm is

$$M_t = H_{t_1}(a + b) - P_t b \text{ inch-pounds,} \qquad (6)$$

and the corresponding stress is

$$K_s = \frac{9\, M_t}{2\, fe^2} \text{ pounds per square inch.} \qquad (7)$$

The total compressive stress should be found by adding the stresses found separately by equations (2), (4) and (5). Thus

$$K_{\text{total}} = K_{b_r} + K_{b_t} + K_c. \qquad (8)$$

The maximum stress may now be found by equation (9) of the last article. This maximum stress should not exceed 10,000 pounds per square inch.

After the above calculations have been made the lengths of the bearings should be determined and the new values of a, a', b, c, c', e and f determined. If these new values differ materially from those assumed for the first calculation, new calculations must be made, using the new dimensions.

When the diameters of the shaft at the various points have been determined, the stresses at bearing 2, at the crank-pin and under the flywheel should be found, using the theorem of 3 moments. This should be done with the shaft in position of maximum torque. The loads are then P' at the center of the crank-pin in the horizontal direction and W under the flywheel in a vertical direction. There is also a twisting moment between bearing 2 and the flywheel equal to $P_t r$. In this work it will not be necessary to take into account the belt tensions.

Fig. 229 shows disposition of loads on a shaft to be used for calculation by the theorem of 3 moments. The equation to be used is

$$M_1L' + 2M_2(L' + L'') + M_3L'' = -P'L'^2(K' - K'^3) - WL''^2(2K'' - 3K''^2 + K''^3),$$

where M_1, M_2 and M_3 are the bending moments at bearings 1, 2 and 3 respectively, L' and L'' are the dimensions shown on Fig. 229, K' is the ratio of a to L' and K'' the ratio of c to L''. This

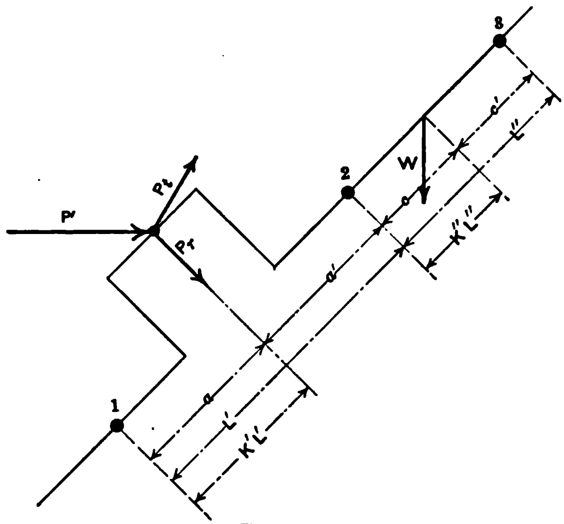


FIG. 229.

equation should be used to find M_2, the bending moment at bearing 2. In solving this equation it is obvious that the bending moments at 1 and 3, M_1 and M_3, are zero. Hence

$$M_{p_2} = -\frac{P'L'^2(K - K^3)}{2(L' + L'')} \text{ inch-pounds}$$

for the piston pressure alone. When using the flywheel weight P' is zero and

$$M_{w_2} = -\frac{WL''^2(2K'' - 3K''^2 + K''^3)}{2(L' + L'')} \text{ inch-pounds.}$$

The reactions should be found for each case so the complete bending moment diagrams may be drawn.

$$R_{p_1} = \frac{M_2}{L'} + P'(1 - K),$$

$$R_{p_3} = \frac{M_2}{L''},$$

$$R_{p_2} = P' - R_{p_1} - R_{p_3}.$$

when P' is considered.

$$R_{w_1} = \frac{M_2}{L'},$$

$$R_{w_2} = WK'' - \frac{M''}{L''},$$

$$R_{w_3} = W - R_{w_2} - R_{w_1},$$

when W is considered. In finding these reactions it is necessary to pay particular attention to the signs.

Since the load P' is horizontal and the load W vertical it will be more convenient to find the bending moments for the two separate and plot them with the twisting moment. Then the resultant of the vertical and horizontal bending moments may be found and combined with the combined bending moment to find the equivalent twisting moment. In order to illustrate the calculations an example is given at the end of this chapter.

Multiple-throw Crank-shafts. — The design of multiple-throw crank-shafts may be based on the principles set forth for the single-throw center- and side-crank types. When the horizontal twin-type engine is used for stationary practice the cranks are overhung and set at 90 degrees from each other and there are only two bearings on the shaft. This makes it unnecessary to use the theorem of 3 moments. The moments should be found and the diagrams drawn for at least two positions, one with one crank on the center and the other crank 90 degrees ahead, the other position with one of the cranks at the position of maximum twisting moment.

Vertical multicylinder engines usually do not have overhung cranks but there is a bearing at each end of the shaft. Since the maximum twisting moment for each crank always occurs within the first one-fifth of the expansion stroke, no two maximum twisting moments can occur at the same time in any of the multicylinder combinations now used. For that reason the diameter of the shaft may be calculated on the basis of one maximum twisting moment as before.

Fig. 230 illustrates a crank shaft for a 4-cylinder vertical marine engine. The inclined position of the middle crank-arm must be regarded as a makeshift only to which recourse must be had when the cylinders are so close together that there is no room for a bearing between. In general this construction should be avoided because the inclined arms are subject to the action of additional

forces which not only increase the stresses but complicate the method of computation. In this case the cylinders "fire" in the order I, IV, II, III.

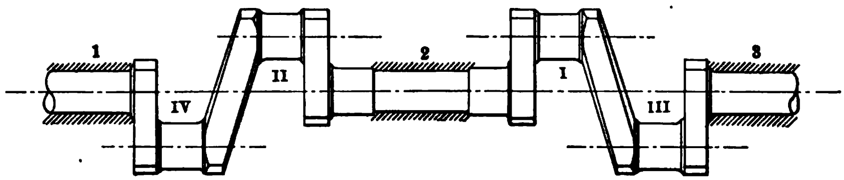


FIG. 230. — Crank-shaft for a Four-cylinder Marine Engine.

The method of procedure in computing a multiple-throw crank-shaft should be as follows:

First, compute dimensions of crank-pins for bearing pressure and strength, find the distances between bearings 1–2 and 2–3 necessary to conform to cylinder dimensions.

Second, assume dimensions of crank-arms.

Third, with shaft in position as shown in Fig. 230, with crank II at the beginning of the explosion stroke, compute reactions at 1 and 2, bending stress in the arm at the left which is toward the load and in the inclined arm at the right.

Fourth, repeat all the calculations under the third item with crank II at the angle of maximum twist and in addition find the twisting in the pin and arms.

Fifth, fix size of bearings to take the load upon them and check stress in them using the theorem of three moments.

If a multicylinder engine has a heavy flywheel the weight should be taken into account if there is no outboard bearing. If there is an outboard bearing the shaft between the end bearing of the engine and the outboard bearing should be designed as a separate beam carrying the flywheel and generator if direct-connected, and belt pulls if belted, in addition to a twisting moment from the engine.

The calculations required above present nothing new except the computation of the stress in the inclined arm. In order to make those clear an example is given here:

Fig. 231 illustrates the forces on the left-hand inclined crank-arm of Fig. 230, crank II on dead center. In this example the shaft will be treated as a simple beam with a load on crank-pin II only. The piston pressure P is 27,000 pounds and the reaction

R is 15,500 pounds. The piston pressure and reaction are resolved parallel and normal to the inclined crank-arm. The components are $P_t = 9200$ pounds, $P_r = 25{,}400$ pounds, $R_t = 5300$

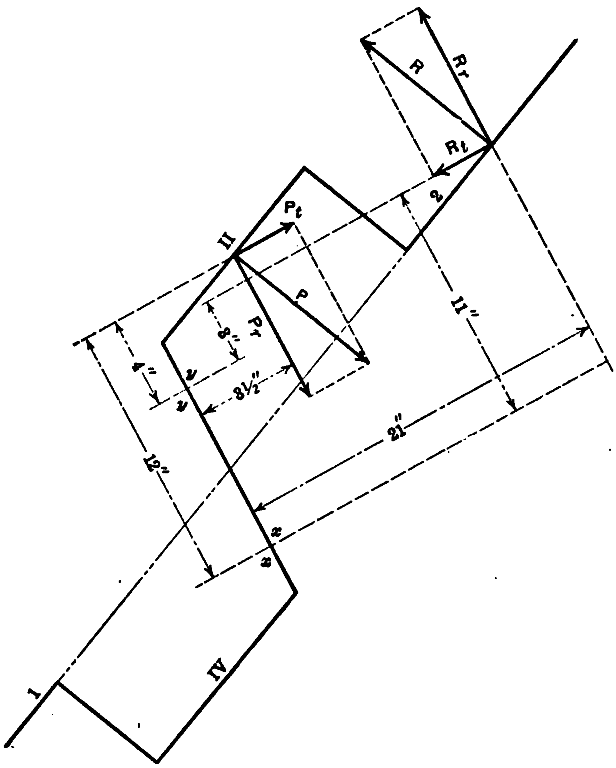


FIG. 231.

pounds and $R_r = 14{,}700$ pounds. The dangerous sections of the arm are at x–x and y–y. The bending moments at x–x are

$$M_{b_1} = 25{,}400 \times 3.5 = 89{,}900 \text{ inch-pounds,}$$

and $$M_{b_2} = 14{,}700 \times 21 = 309{,}000 \text{ inch-pounds.}$$

Since these are in opposite directions they must be subtracted, giving the net bending moment due to the parallel components

$$M_{b_r} = 309{,}000 - 89{,}900 = 219{,}100 \text{ inch-pounds.}$$

The bending moments due to the normal components are

$$M_{b_3} = 9200 \times 12 = 110{,}000 \text{ inch-pounds,}$$

and $$M_{b_4} = 5300 \times 11 = 58{,}000 \text{ inch-pounds.}$$

These also are in the opposite direction and must be subtracted.

$$M_{b_t} = 110{,}000 - 58{,}000 = 52{,}000 \text{ inch-pounds.}$$

It is evident from the figure that M_{b_r} and M_{b_t} are opposite in sign and hence the result is the difference. That is

$$M_{b_{net}} = 219{,}100 - 52{,}000 = 167{,}100 \text{ inch-pounds.}$$

The size of the arm is $6\frac{3}{4} \times 3\frac{13}{16}$ inches. The section modulus is then $\frac{1}{6} \times 6.75 \times \overline{3.8125}^2 = 16.3$ inches^3. The stress in the arm due to the radial load will then be

$$K_{b_x} = \frac{167{,}100}{16.3} = 10{,}250 \text{ pounds per square inch.}$$

The stress at the section *y–y* may be found in the same manner. The bending moments due to P_r and R_r will be the same as at the section *x–x*, the net moment being

$$M_{b_r} = 219{,}100 \text{ inch-pounds.}$$

The moment due to P_t is

$$M_{b_3} = 9200 \times 4 = 36{,}800 \text{ inch-pounds.}$$

That due to R_t is

$$M_{b_4} = 5300 \times 3.125 = 16{,}600 \text{ inch-pounds.}$$

The difference of these is

$$M_{b_t} = 38{,}600 - 16{,}600 = 22{,}000 \text{ inch-pounds.}$$

The net bending moment is

$$M_{b_{net}} = 219{,}100 - 22{,}000 = 197{,}100 \text{ inch-pounds.}$$

The stress is

$$K_{b_y} = \frac{197{,}100}{16.3} = 12{,}100 \text{ pounds per square inch.}$$

Additional Loads on Shaft. — When the gas-engine drives an electric generator the weight of the rotor should be included as a load in the vertical plane when the rotor is between the outboard-bearing and the engine frame. In some instances the generator is driven by means of a flexible coupling and therefore does not cause any bending stress in the engine shaft. If such types of engines as are shown in Fig. 226, (*a*) and (*b*), were direct-connected to generators, the generator rotor would be supported on two in-

dependent bearings and driven by a flexible coupling. This would throw no additional load on the engine shaft but would actually relieve the shaft of one of its loads, the belt tensions. If the type shown at (*c*) is direct-connected to a generator then the weight of the rotor would come on the right-hand main-bearing and the outboard-bearing, causing additional stresses in the shaft.

Still another load that must be accounted for is the magnetic pull on the rotor due to the displacement of the shaft from the center-line of the stator of the generator. This displacement is due to two causes, the deflection of the shaft caused by the loads on it and the displacement due to wear in the bearings. After the total displacement is found the magnetic pull may be found after the type of generator is known. This pull must be treated as additional weight at the center-line of the rotor.

Deflection of the Shaft. — If W' is the sum of the loads concentrated at the center of the flywheel the deflection of the shaft will be

$$\frac{W'c^2c'^2}{3\,IEm}\text{ inches,}$$

where c, c' and m are dimensions given in Fig. 227, I is the plane moment of inertia of the shaft, E is the modulus of elasticity and W' is the combined load due to the flywheel, rotor and magnetic pull. The allowable deflection depends on the type of generator but should not exceed 0.03 inch under any conditions.

Size of Journals. — The bearing pressure in the main-bearings is determined by the piston pressure, the weight of the flywheel and the belt pulls. In case of a direct-connected machine the weight of the rotor and the magnetic pull must be considered. The pressure on the bearings due to the gas pressure in the cylinder is not in the same direction as the pressure due to the flywheel weight in the horizontal engine. In vertical engines the pressures are all vertical and should be added and it is customary to add them in the horizontal engine also. The main-bearing on the side from which the power is taken off is the one most heavily loaded. The length of the bearing depends upon the rapidity with which the heat of friction may be dissipated. Hence the length depends upon the mean pressure rather than upon the maximum pressure. This is not the mean effective pressure but the mean pressure measured from the atmospheric

line of the indicator card for each stroke. The inertia of the moving parts must also be considered on the suction and exhaust strokes of a single-acting four-cycle engine. During the suction and exhaust strokes the mean pressure may be regarded as the mean height of the inertia curve regardless of whether it is positive or negative, as the suction or exhaust pressure would be added to the inertia for half the stroke and subtracted for the other half. If the mean pressure on the four strokes should be as follows:

Admission,	$p_1 =$	7	pounds	per	square	inch,
Compression,	$p_2 =$	24	"	"	"	"
Expansion,	$p_3 =$	110	"	"	"	"
Exhaust,	$p_4 =$	7	"	"	"	"
Total		148	"	"	"	"

the mean pressure would be 37 pounds per square inch $= p_m$. Then the total mean load due to the cylinder pressure on the main bearing at the right, Fig. 227, would be

$$\frac{P_m}{2} = \frac{p_m \times \text{area piston}}{2}. \tag{1}$$

The total load causing friction would be

$$Q = \frac{P_m}{2} + V_2, \tag{2}$$

where V_2 is the reaction due to all loads between bearings 2 and 3. The work of friction per second per square inch of projected area of the bearing would be

$$F_2 = \frac{Q\pi\, d_2 N \times 0.05}{d_2 \times l_2 \times 12 \times 60} \text{ foot-pounds per second,} \tag{3}$$

where 0.05 is the coefficient of friction and d_2 and l_2 are the diameter and length respectively of the bearing at 2 in inches. This should not exceed 70 foot-pounds per square inch of projected area per second. The same computation should be made for the crank-pin which carries only the total pressure due to the piston pressure. Then

$$F = \frac{P_m \pi\, dN \times 0.05}{d \times l \times 12 \times 60} \text{ foot-pounds per second,} \tag{4}$$

where d is the diameter and l the length of the crank-pin in inches. This value should not exceed 60 foot-pounds per square inch per second.

The pressure per square inch on the crank-pin due to the explosion pressure and on the main-bearing due to the explosion pressure and the reaction of weight s and rope drive should be found. These values should not exceed 1500 pounds per square inch for the crank-pin and 600 pounds per square inch for the main-bearing.

The length of the bearing at 2, Fig. 227, should be found from equation (3) by putting in the value of 70 for F. Then every other term of the equation is known and l_2 may be found. This value of l_2 should then be used to find new values of a, c, etc., in Fig. 227 and a new diameter of shaft computed if the values of a, c, etc., differ greatly from the original assumed values.

It will be noticed that d_2 and d in equations (3) and (4) cancel. This leaves l_2 and l as the only unknowns in these equations which may be used to find the approximate length of bearing and crank-pin at the beginning of the shaft calculations.

Main-bearings. — The main-bearings of an engine are as important as the shaft and should be constructed with the same care given the shaft. There is nothing about the bearings that can be designed rationally after the diameter and length have been computed. The general types of bearings in use are:

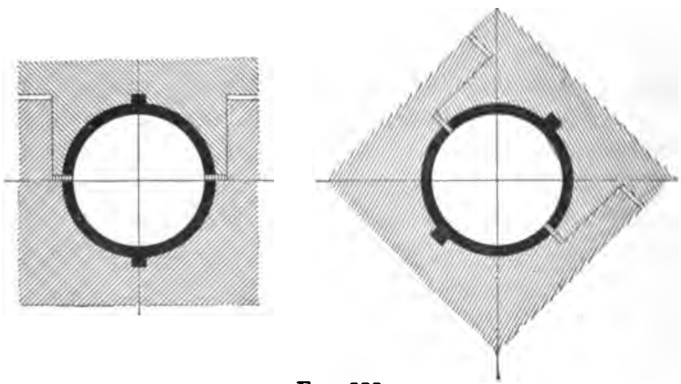


Fig. 232.

Bearings with horizontal or inclined cap such as are illustrated in Fig. 232. The cap is adjustable by means of shims to take up wear. This form of bearing is used only in small engines.

Self-lubricated bearings where lubrication is secured by means of an oil-ring as in Figs. 233 and 234. The latter is a bearing used on the Mietz and Wiess oil-engine. This bearing is bolted

to the frame of the engine. The figure also illustrates an efficient system of crank-pin lubrication.

Four-part bearings with wedge adjustment similar to those used in steam-engine practice. This type of bearing is illustrated

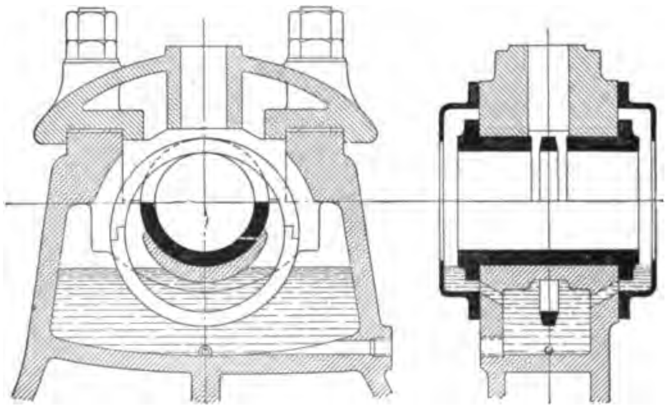


FIG. 233. — Ring-oiled Bearing.

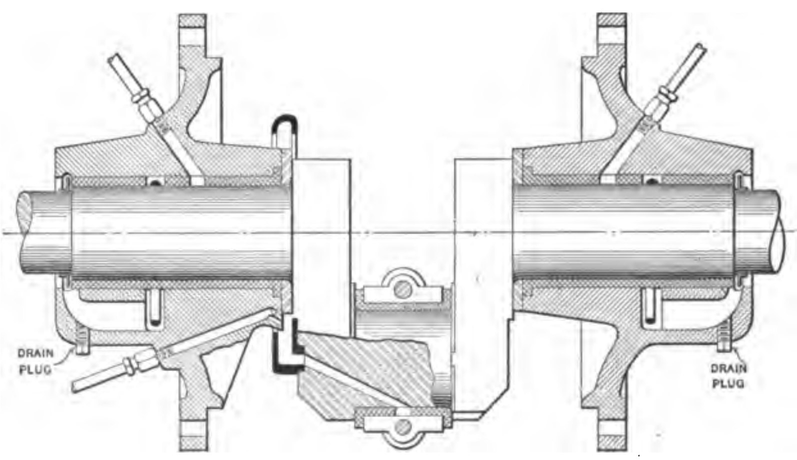


FIG. 234. — Mietz and Weiss Crank-shaft Bearings.

in Fig. 235 which shows a bearing for a large double-acting gas-engine. This particular bearing has two wedges for horizontal adjusting. In many cases there is only one wedge fitted for this purpose. Vertical adjustment is taken care of by shims. This bearing also illustrates the water-cooling principle applied to large bearings.

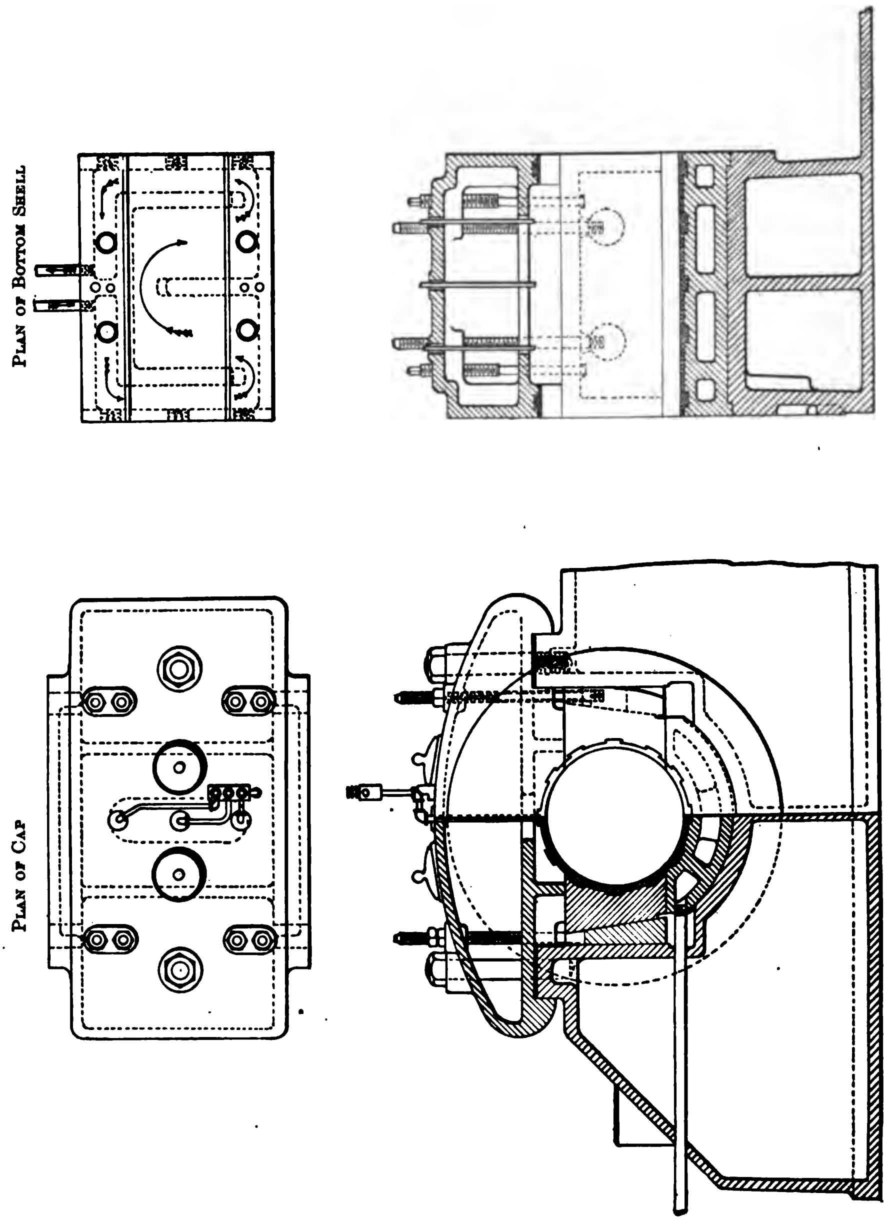


FIG. 235.—Four-part Bearing for Large Gas-engine.

The bed-plate, bearings and shaft for a two-cylinder four-cycle, single-acting Diesel engine are shown in Fig. 236.

The material used for lining the shells of crank-shaft bearings

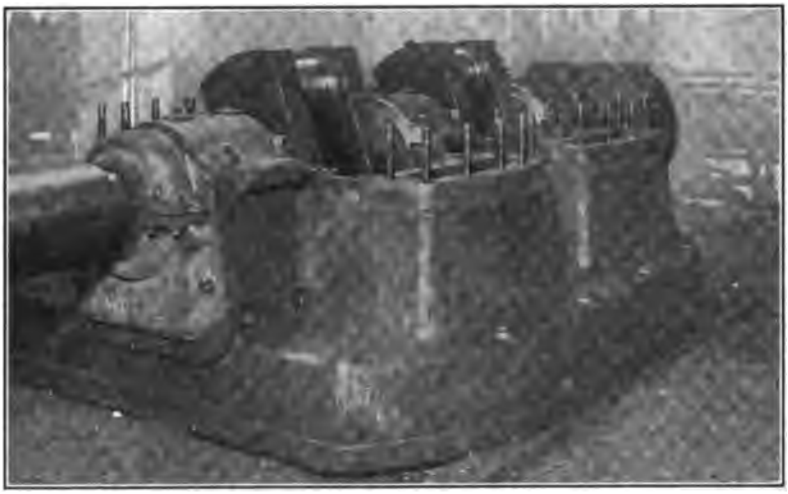


FIG. 236. — Bearings and Crank-shaft for a Vertical Diesel Engine.

is almost always babbitt metal. The shells themselves are made of cast-iron or cast-steel. The thickness of the babbitt depends on the diameter of the shaft, varying from $\frac{3}{16}$ to $\frac{1}{2}$ inch. The babbitt is held in place by means of dove-tail grooves, as shown

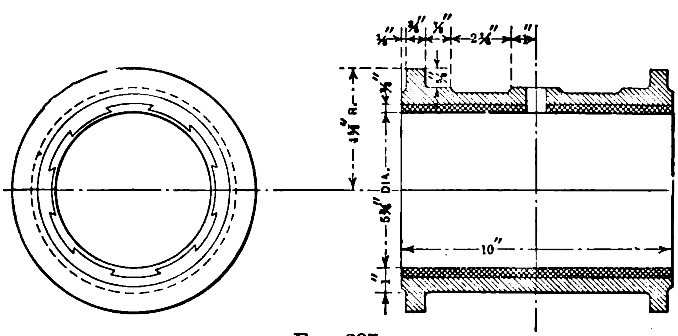


FIG. 237.

in Fig. 237 which illustrates the bearing shells for a $5\frac{3}{4}$-inch shaft for a 10 × 18-inch horizontal oil engine.

Example. — In order to illustrate the methods of shaft calculation used in this chapter there will be made the calculations for

a shaft for a 16 × 24-inch single-acting, four-cycle single-cylinder engine, 200 r.p.m. The mean effective pressure to be 70 pounds per square inch, maximum or explosion pressure 375 pounds per square inch and maximum brake horse-power 72. The type of shaft used will be that shown in Fig. 227. The width of belt required will be about 11 inches with a thickness of $\frac{7}{32}$ inch, making the face of the flywheel 12 inches, the diameter of the wheel being about 9 feet. The tension in the tight side of the belt will be

$$T_1 = \tfrac{7}{32} \times 11 \times 400 = 960 \text{ pounds,} \tag{1}$$

where 400 is the working tension per square inch of belt. The difference in tensions will be

$$T_1 - T_2 = \frac{72 \times 550}{90} = 440 \text{ pounds,} \tag{2}$$

where 550 = foot-pounds per second in one horse-power and 90 is the velocity of the belt in feet per second. From (1) and (2) we get $T_2 = 520$ pounds. The sum of the tensions will then be

$$T_1 + T_2 = 960 + 520 = 1480 \text{ pounds.} \tag{3}$$

The total weight of the flywheel W we will assume to be 12,000 for use in this problem.

Crank-pin. — The distance n may be assumed to be twice the cylinder diameter or 32 inches. The total piston pressure with the crank on the inner center is

$$P = \frac{\pi D^2}{4} \times 375 = 75{,}400 \text{ pounds.} \tag{4}$$

Assuming a and a' equal, the reactions H_1 and H_2 will be equal, 37,700 pounds each. The bending moment at the center of the crank-pin will then be

$$M_b = 37{,}700 \times 16 = 603{,}200 \text{ inch-pounds.}$$

The diameter will be

$$d = \sqrt[3]{\frac{603{,}200 \times 32}{12{,}000 \times \pi}} = 8 \text{ inches.} \tag{5}$$

Allowing 1500 pounds pressure per square inch of projected area, based on the explosion pressure, we find the length of the crank-pin to be

$$l = \frac{75{,}400}{8 \times 1450} = 6\tfrac{1}{2} \text{ inches.} \tag{6}$$

Stress in Left-hand Crank-arm. — The thickness of the crank-arms will be

$$e = 0.65 \times 7.75 + 0.25 = 5\tfrac{1}{4} \text{ inches.} \tag{7}$$

The width will be

$$f = 1.125 \times 7.75 + 0.5 = 9\tfrac{1}{4} \text{ inches.} \tag{8}$$

The distance b is then $\frac{6.5}{2} + \frac{5.25}{2} = 5\tfrac{7}{8}$ inches and the distance $a - b$ will be $10\tfrac{1}{8}$ inches. The bending moment on the left-hand arm is then

$$M_b = 37{,}700 \times 10.125 = 382{,}000 \text{ inch-pounds,}$$

and the stress will be

$$K_b = \frac{6 \times 382{,}000}{5.25 \times 5.25 \times 9.25} = 9000 \text{ pounds per square inch.} \tag{9}$$

The stress in the **right-hand crank-arm** will be the same.

Shaft Under Flywheel. — Before the moments under the flywheel may be computed it is necessary to find c, c', the length of bearing $2 = l_2$, and the length of bearing $3 = l_3$. In this type of shaft we may assume the length of all the main-bearings to be the same. In this case $l_2 = \left(a' - \frac{l}{2} - e\right) 2 = 15$ inches. The face of the flywheel is 12 inches. The bending moment due to the weight of the flywheel will be

$$M_w = \frac{12{,}000}{2} \times 21 = 126{,}000 \text{ inch-pounds}$$

and that due to the belt pull will be

$$M_b = \frac{1480}{2} \times 21 = 15{,}120 \text{ inch-pounds.}$$

The combined bending moment will be

$$M_{\text{total}} = \sqrt{\overline{126{,}000}^2 + \overline{15{,}120}^2} = 127{,}000 \text{ inch-pounds.}$$

The diameter to correspond will be

$$d_w = \sqrt[3]{\frac{32 \times 127{,}000}{\pi \times 6000}} = 6 \text{ inches.} \tag{10}$$

Crank at Angle of Maximum Twisting Moment. — It was found by graphical methods that for a length of connecting-rod $5\tfrac{1}{2}$ times the crank radius the force along the rod at the time maximum twisting occurs is 56,800. Resolving this force parallel

and normal to the crank we find $P_r = 45{,}400$ pounds and $P_t = 38{,}200$ pounds. Referring to Fig. 228 we find

$$\begin{aligned} H_{t_1} &= H_{t_4} = 19{,}100 \text{ pounds,} \\ H_{r_1} &= H_{r_4} = 22{,}700 \text{ pounds,} \\ V_2 &= V_3 = 6000 \text{ pounds.} \end{aligned}$$

Shaft Under Flywheel. — The combined bending moment under the flywheel is the same as before, 127,000 inch-pounds. The twisting moment is

$$M_t = 38{,}200 \times 12 = 458{,}000 \text{ inch-pounds.}$$

If this is combined with the bending moment by Guest's law the equivalent twisting moment will be

$$M_{t_e} = \sqrt{\overline{458{,}000}^2 + \overline{127{,}000}^2} = 473{,}000 \text{ inch-pounds.}$$

The diameter to correspond is

$$d_w = \sqrt[3]{\frac{16 \times 473{,}000}{\pi \times 4000}} = 8\tfrac{1}{2} \text{ inches.} \tag{11}$$

Shaft at Juncture of Right-hand Crank-arm. — The twisting moment here is the same as that under the flywheel. The bending moment is

$$M_b = \frac{56{,}800}{2}\left(16 + 5\tfrac{7}{8} + \frac{5.25}{2}\right) - 56{,}800\left(5\tfrac{7}{8} + \frac{5.25}{2}\right)$$
$$= 215{,}700 \text{ inch-pounds.}$$

The equivalent twisting moment is

$$M_{t_e} = \sqrt{\overline{458{,}000}^2 + \overline{215{,}700}^2} = 507{,}000 \text{ inch-pounds.}$$

The diameter to correspond is

$$d_2 = \sqrt[3]{\frac{16 \times 507{,}000}{\pi \times 6000}} = 7\tfrac{1}{2} \text{ inches.} \tag{12}$$

The stress used here is higher than that used under the flywheel for two reasons; the condition of loading which was assumed at this point is more severe than the actual condition, and the deflection under the flywheel must be kept small.

Crank-pin. — The bending moment is

$$M_b = \frac{56{,}800}{2} \times 16 = 455{,}000 \text{ inch-pounds,}$$

and the twisting moment is

$$M_t = 19{,}100 \times 12 = 229{,}000 \text{ inch-pounds.}$$

The equivalent twisting moment is

$$M_{t_e} = \sqrt{\overline{455,000}^2 + \overline{229,000}^2} = 511,000 \text{ inch-pounds.}$$

The diameter to correspond is

$$d = \sqrt[3]{\frac{511,000 \times 16}{\pi \times 5000}} = 8 \text{ inches.} \tag{11}$$

Left-hand Crank-arm at Juncture of Pin. — The bending moment due to the radial component is

$$M_{b_r} = 22,700 \times 10.125 = 230,000 \text{ inch-pounds.}$$

The stress to correspond is

$$K_{b_r} = \frac{6 \times 230,000}{5.25 \times 5.25 \times 9.25} = 5420 \text{ pounds per square inch.}$$

The bending moment due to the tangential component is

$$M_{b_t} = 19,100\,(12 - 4) = 152,800 \text{ inch-pounds.}$$

The stress to correspond is

$$K_{b_t} = \frac{6 \times 152,800}{5.25 \times 9.25 \times 9.25} = 2050 \text{ pounds per square inch.}$$

The stress in direct compression is

$$K_c = \frac{22,700}{5.25 \times 9.25} = 430 \text{ pounds per square inch.}$$

The total compressive stress is

$$K_{c_{\text{total}}} = 5420 + 2050 + 430 = 7900 \text{ pounds per square inch.}$$

The twisting moment on the arm is

$$M_t = 19,100 \times 10.125 = 193,500 \text{ inch-pounds,}$$

and the stress to correspond is.

$$K_s = \frac{9 \times 193,500}{2 \times 9.25 \times 5.25 \times 5.25} = 3420 \text{ pounds per square inch.}$$

The total combined stress in the arm will be

$$K_{\max} = \frac{7900}{2} + \sqrt{\frac{\overline{7900}^2}{4} + \overline{3420}^2} = 9200 \text{ pounds per square inch.} \tag{12}$$

Right-hand Crank-arm. — The bending moment due to the radial component is

$$M_{b_r} = 22,700 \times 21.875 - 45,400 \times 5.875 = 232,000 \text{ inch-pounds.}$$

The stress to correspond is

$$K_{b_r} = \frac{6 \times 232{,}000}{5.25 \times 5.25 \times 9.25} = 5460 \text{ pounds per square inch.}$$

The bending moment due to the tangential component is a maximum where the arm joins the shaft and is

$$M_{b_t} = 38{,}200\,(12 - 3.75) + 19{,}100 \times 3.75 = 386{,}600 \text{ inch-pounds.}$$

The stress to correspond is

$$K_{b_t} = \frac{6 \times 386{,}600}{5.25 \times 9.25 \times 9.25} = 5160 \text{ pounds per square inch.}$$

The direct stress in compression is the same as in the other arm, 430 pounds per square inch. The total compressive stress on one corner is

$$K_{c_{total}} = 5460 + 5160 + 430 = 11{,}050 \text{ pounds per square inch.}$$

The twisting moment on the arm is

$$M_t = 19{,}100 \times 21.875 - 38{,}200 \times 5.875 = 196{,}000 \text{ inch-pounds.}$$

The twisting stress is

$$K_s = \frac{9 \times 196{,}000}{2 \times 9.25 \times 5.25 \times 5.25} = 3460 \text{ pounds per square inch.}$$

The total combined stress in the arm will be

$$\left. K_{max} = \frac{11{,}050}{2} + \sqrt{\frac{\overline{11{,}050}^2}{4} + \overline{3460}^2} = 12{,}045 \text{ pounds per square inch.} \right\} \qquad (13)$$

Length of Bearing at 2. — The bearing at 2 has the largest total reactions, hence this particular one should be designed for bearing pressure. The mean reactions there are due to the mean pressure exerted by the pressure in the cylinder and the weight of reciprocating parts, one-half the flywheel weight and one-half the belt pull. The mean reaction due to the piston pressure and reciprocating parts should be found from the indicator card and the inertia diagram for each stroke and the mean of the four used. In this case we will assume the mean pressure of the four strokes of the cycle to be 40 pounds per square inch of piston. The total reaction at 2 will then be

$$Q_2 = \frac{40 \times 201 + 1480 + 12{,}000}{2} = 10{,}760 \text{ pounds.}$$

The diameter of the shaft at this point, bearing 2, was found to be $7\frac{1}{2}$ inches, equation (12). From the length of crank-pin and thickness of arm we may find the length of bearing 2. Thus

$$\frac{l_2}{2} = a' - \frac{1}{2} - e = 16 - \frac{6.5}{2} - 5.25 = 7\tfrac{1}{2} \text{ inches,}$$

from which we find the total length of l_2 to be 15 inches. The work of friction per square inch of projected area per second is

$$F = \frac{10{,}760 \times \pi \times 7.5 \times 200 \times 0.05}{7.5 \times 15 \times 12 \times 60} = 31.3 \text{ foot-pounds.}$$

As this value is far below the maximum value of 70 we will investigate the total pressure on bearing 2. This is

$$R_2 = \frac{201 \times 375 + 1480 + 12{,}000}{2 \times 7.5 \times 15} = 395 \text{ pounds per square inch.}$$

Since this value may run up to 600 pounds per square inch and since the work of friction is low the length l_2 may be reduced. If we make $l_2 = 12$ inches then a', Fig. 228, will be

$$a' = \frac{6.5}{2} + 5.25 + 6 = 14\tfrac{1}{4} \text{ inches}$$

instead of 16 inches as was first assumed. The calculations should be gone over again in exactly the same manner and new diameters and new stresses found as before. In order to avoid tiresome repetition this recalculation is omitted here.

Bending Moment in Bearing 2. — Using the dimensions as in the previous calculations we may find the bending at bearing 2 by the theorem of three moments. Referring to Fig. 229 the values of K' and K'' will each be 0.5. The load P' between 1 and 2 will be 56,800 which we may assume to be in a horizontal direction. The load W between bearings 2 and 3 is 12,000 pounds, neglecting the belt pulls. The bending moment at bearing 2 due to P' is

$$M_{2p} = -\frac{56{,}800\,(0.5 - 0.5^3)\,\overline{32}^2}{2\,(32 + 42)} = -147{,}500 \text{ inch-pounds,}$$

$$R_{p_1} = -\frac{147{,}500}{32} + 56{,}800\,(1 - 0.5) = 23{,}780 \text{ pounds,}$$

$$R_{p_3} = -\frac{147{,}500}{42} = -3510 \text{ pounds,}$$

$$R_{p_2} = 56{,}800 - 23{,}780 + 3510 = 36{,}530 \text{ pounds.}$$

The bending moment at bearing 2 due to W is

$$M_{2w} = -\frac{12,000 \times \overline{42}^2 (2 \times 0.5 - 3 \times 0.25 + 0.125)}{2\,(32 + 42)}$$
$$= -53,700 \text{ inch-pounds,}$$
$$R_{w_1} = -\frac{53,700}{32} = -1675 \text{ pounds,}$$
$$R_{w_2} = 12,000 \times 0.5 - \left(-\frac{53,700}{42}\right) = 7280 \text{ pounds,}$$
$$R_{w_3} = 12,000 - 7280 + 1675 = 6395 \text{ pounds.}$$

The twisting moment on the shaft is

$$M_t = 38,200 \times 12 = 458,000 \text{ inch-pounds.}$$

The bending moment at the center of the crank-pin due to P' is

$$M_p = R_{p_1} \times a = 23,780 \times 16 = 380,000 \text{ inch-pounds.}$$

The bending moment under W due to W is

$$M_w = R_{w_3} \times c_1 = 7280 \times 21 = 153,000 \text{ inch-pounds.}$$

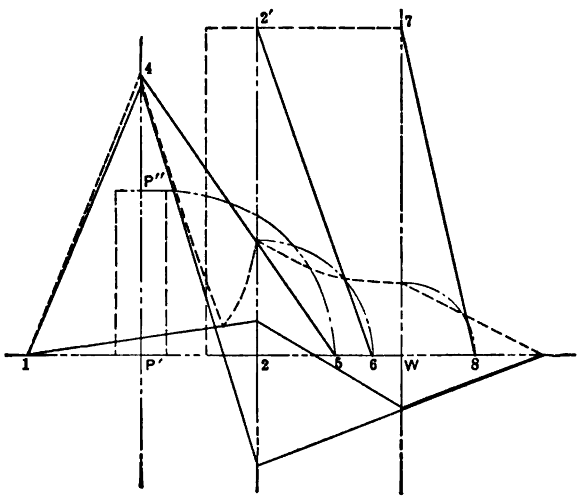


FIG. 238.

The bending moment, twisting and combined diagrams are shown in Fig. 238. The moments due to the piston pressure and fly-wheel being at right angles to each other are combined by finding the square root of the sum of their squares. The combined bending moments are shown by the dotted broken line. At the

center of the crank-pin P', at bearing 2 and at the center of the flywheel W the combined bending moments are combined with the twisting moments at those points, resulting in equivalent twisting moments 4–5 at the crank-pin, 2′–6 at the center of bearing 2 and 7–8 at the center of the flywheel. These moments have the following values:

$$4\text{–}5 = 469{,}000 \text{ inch-pounds,}$$
$$2'\text{–}6 = 483{,}000 \text{ inch-pounds,}$$
$$7\text{–}8 = 466{,}000 \text{ inch-pounds.}$$

In the computations for this shaft as a simple beam it was found that the combined bending and twisting moment at the center of the crank-pin was 511,000 inch-pounds, that at the place where bearing 2 joins the crank-arm was 507,000 inch-pounds and that under the flywheel was 473,000 inch-pounds. Thus the shaft is safe as already designed.

Five-bearing Crank-shaft. — The five-bearing crank-shaft for a four-cylinder vertical engine with an overhung flywheel is shown in Fig. 239. This shaft should be investigated for at least 3 conditions of loading as follows:

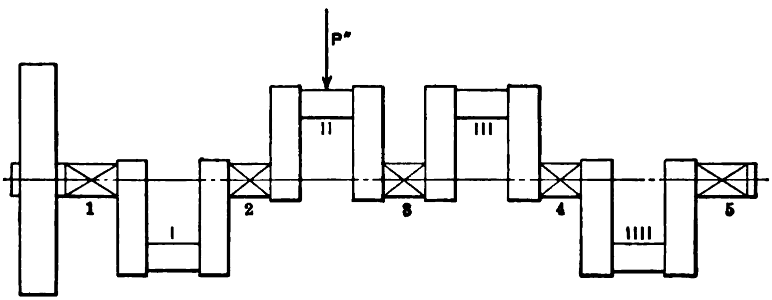


FIG. 239.

First, with the second crank from the left on the top center at the beginning of the explosion stroke, assuming the engine to be just starting up. Then the load on this crank is the explosion pressure multiplied by the area of the piston and the loads on the other cranks are zero.

Second, with the same crank at the angle of maximum twisting moment. In this position the load on this crank should be found as in the previous problem. The loads on the other cranks will be practically zero.

Third, with the cranks in position given under the first condition but with the engine up to speed. The load P'', Fig. 240, will be the explosion pressure minus the force of inertia, multiplied by the area of the piston. The other loads will be the inertia in each case multiplied by the area of the piston. P''' will be upward, P' and P'''' will be downward.

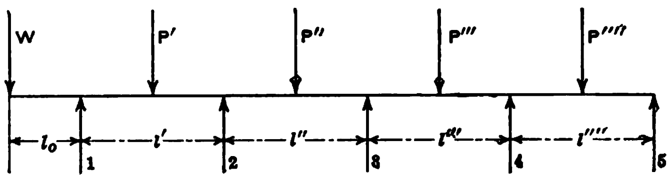


FIG. 240.

In all of the above cases the bending moments should be found by the theorem of three moments and plotted as was done for the three-bearing shaft. In the second case there will be twisting in addition to the bending.

The moments may be found by the following equations. The spans are all the same length except l_0 and the loads are in the middle of the spans in every case except the flywheel which is an overhung load. The subscripts of the bending moments refer to the bearing numbers.

$$M_1 = -Wl_0,$$

$$M_1 l' + 2(l' + l'')M_2 + M_3 l'' = -P' l'^2 (\tfrac{1}{2} - \tfrac{1}{8}) - P'' l''^2 (1 - \tfrac{3}{4} + \tfrac{1}{8}),$$

$$M_2 l'' + 2(l'' + l''')M_3 + M_4 l''' = -P'' l''^2 (\tfrac{1}{2} - \tfrac{1}{8}) - P''' l'''^2 (1 - \tfrac{3}{4} + \tfrac{1}{8}),$$

$$M_3 l''' + 2(l''' + l'''')M_4 + M_5 l'''' = -P''' l'''^2 (\tfrac{1}{2} - \tfrac{1}{8}) - P'''' l''''^2 (1 - \tfrac{3}{4} + \tfrac{1}{8}),$$

$$M_5 = 0.$$

The shear at the right of the various supports will be as follows:

$$V_1 = \frac{M_2 - M_1}{l'} + \frac{P'}{2},$$

$$V_2 = \frac{M_3 - M_2}{l''} + \frac{P''}{2},$$

$$V_3 = \frac{M_4 - M_3}{l'''} + \frac{P'''}{2},$$

$$V_4 = \frac{M_5 - M_4}{l''''} + \frac{P''''}{2}.$$

The moments under the loads will then be

$$M' = M_1 + \frac{V_1 l'}{2},$$

$$M'' = M_2 + \frac{V_2 l''}{2},$$

$$M''' = M_3 + \frac{V_3 l'''}{2},$$

$$M'''' = M_4 + \frac{V_4 l''''}{2}.$$

EXERCISES

1. Design the crank-arm for a 16 × 26 gas-engine, explosion pressure 375 lbs. per sq. in. Referring to Fig. 222 assume $l_1 = 7$ ins., diameter of shaft at bearing 1 = 8 ins. Maximum twisting moment on shaft occurs at a crank angle of 35 degrees at which time there is a pressure on the piston of 125 lbs. per sq. in. The connecting-rod of the engine is 65 ins. long.

2. Find the bending moments at bearing 1 and under the flywheel of the above engine when the crank is on the dead center. Assume $c = c' = 23$ ins., $W = 20{,}000$ lbs., $l = 16$ ins., $e = 6$ ins. and $T_1 + T_2 = 3000$ lbs.

3. Find the bending and twisting moments at 1 and 2 when the crank is in the position of maximum twisting moment as given in Prob. 1.

4. Referring to Fig. 227 assume $P = 80{,}000$ lbs., $r = 11$ ins., $a = a' = 16$ ins., $b = 9$ ins., $e = 5\frac{1}{2}$ ins., $f = 9$ ins., $W = 14{,}000$ lbs., $c = 22$ ins., $c' = 18$ ins. and $T_1 + T_2 = 2500$ lbs. Find H_1, H_2, V_2, and V_3. Find the bending moments at the center of the crank-pin and at W.

5. Using the data given in Prob. 4 find the stress in the left-hand crank-arm when the crank is at an angle of 35 degrees, pressure on the piston 135 lbs. per sq. in. Length of connecting-rod 60 ins. Dia. cyl. = 16 ins.

6. Using the conditions given in Probs. 4 and 5 find the diameter of the crank-pin.

7. Using the conditions given in Probs. 4 and 5 find the diameter of the shaft under the flywheel.

8. Referring to Fig. 240, assume $l' = l'' = l''' = l'''' = 16$ ins., $l_0 = 12$ ins., $W = 2000$ lbs, $P' = 1000$, $P'' = 16{,}000$, $P''' = 1200$ and $P'''' = 1500$ lbs. Find the bending moments at bearings 1, 2, 3, 4 and 5 and at each load. Draw the bending moment diagram.

INDEX

TABLES